



# **FEASIBILITY STUDY FOR CONVERTIBLE ENGINE TORQUE CONVERTER**

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**ALLISON GAS TURBINE DIVISION  
GENERAL MOTORS CORPORATION**

Prepared For

**NATIONAL AERONAUTICS  
AND  
SPACE ADMINISTRATION**

**NASA LEWIS RESEARCH CENTER  
CONTRACT NAS3-24092**

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**OCTOBER 1985**

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## I. SUMMARY

NASA Lewis Research Center (LeRC) sponsored the Feasibility Study for a Convertible Engine Torque Converter under NASA contract NAS3-24092 to Allison Gas Turbine Division of General Motors Corporation (GMC). Allison was assisted in the study by other GMC centers, namely Detroit Diesel Allison Division (DDA) and the Advanced Engineering Staff (AES) of the General Motors Technical Staff. The torque converter application in a convertible turbofan/turboshaft engine was recommended for concept feasibility evaluation in the Rotorcraft Convertible Engine study performed under NASA contract NAS3-22742 by Allison.

The feasibility study has shown that a dump/fill type torque converter has excellent potential for the convertible fan/shaft engine. The torque converter space requirement permits internal housing within the normal flow path of a turbofan engine at acceptable engine weight. The unit permits operating the engine in the turboshaft mode by decoupling the fan. To convert to turbofan mode, the torque converter overdrive capability brings the fan speed up to the power turbine speed to permit engagement of a mechanical lockup device when the shaft speeds are synchronized. The conversion to turbofan mode can be made without droop of power turbine speed in less than 10 sec. Total thrust delivered to the airplane by the prop rotor, fan, and engine during the transient can be controlled to prevent loss of air speed or altitude. Heat rejection to the oil is low, and additional oil cooling capacity is not required. The turbofan engine aerodynamic design is basically uncompromised by convertibility and allows proper fan design for quiet and efficient cruise operation. Although the core shaft engine is not supercharged, turbofan engines sized for high speed rotorcraft can have adequate core size to provide shaft power in the helicopter mode without supercharging.

Although the results of the feasibility study are exceedingly encouraging, it must be noted that they are based on extrapolation of limited existing data on torque converters. For example, the tip speeds of the rotating parts will be increased by nearly a factor of five in the aircraft application. Thus potential problems with the torque converter must be explored in experimental test programs. Cavitation, which exists in current state-of-the-art torque converters, will be more severe at the conditions imposed by the turbine engine application. The effects of cavitation on performance, noise, and erosion need to be explored at various levels of charging pressure and cavity pressure. High axial forces can be produced by reaction between the turbine hub and pump elements of the torque converter, but these may be reduced to acceptable levels by vanes inducing static pressure reduction. The dump/fill operation of the torque converter, while done successfully on automotive units, requires careful control of oil entry and exit to avoid fluid unbalance at the high rotative speeds encountered in turbine engine application. A component test program with three trial torque converter designs and concurrent computer modeling for fluid flow, stress, and dynamics, updated with test results from each unit, is recommended.

## II. INTRODUCTION

The convertible engine torque converter provides a means to decouple the fan in a turbofan engine and thus make gas generator output available to turn a shaft. The resulting convertible engine is a turbofan/turboshaft engine capable of delivering thrust or shaft horsepower. Its chief application is for high speed rotorcraft, such as the fold tilt rotor, where shaft power is used to drive a prop rotor that can be tilted to provide rotor lift in the helicopter mode or propeller thrust in the wing-borne flight mode. Turbofan engine thrust is needed for propulsion when the aircraft is in the high speed configuration with the prop rotors folded. The convertible engine provides both functions at much less weight and installation complexity than separate turboshaft and turbofan engines.

The objectives of the Feasibility Study for Convertible Engine Torque Converter program were to prepare a conceptual design of a gas turbine housed, flight weight torque converter for primary use in a convertible fan shaft engine and to define a work plan to solve problems identified in the study that require research and technology effort.

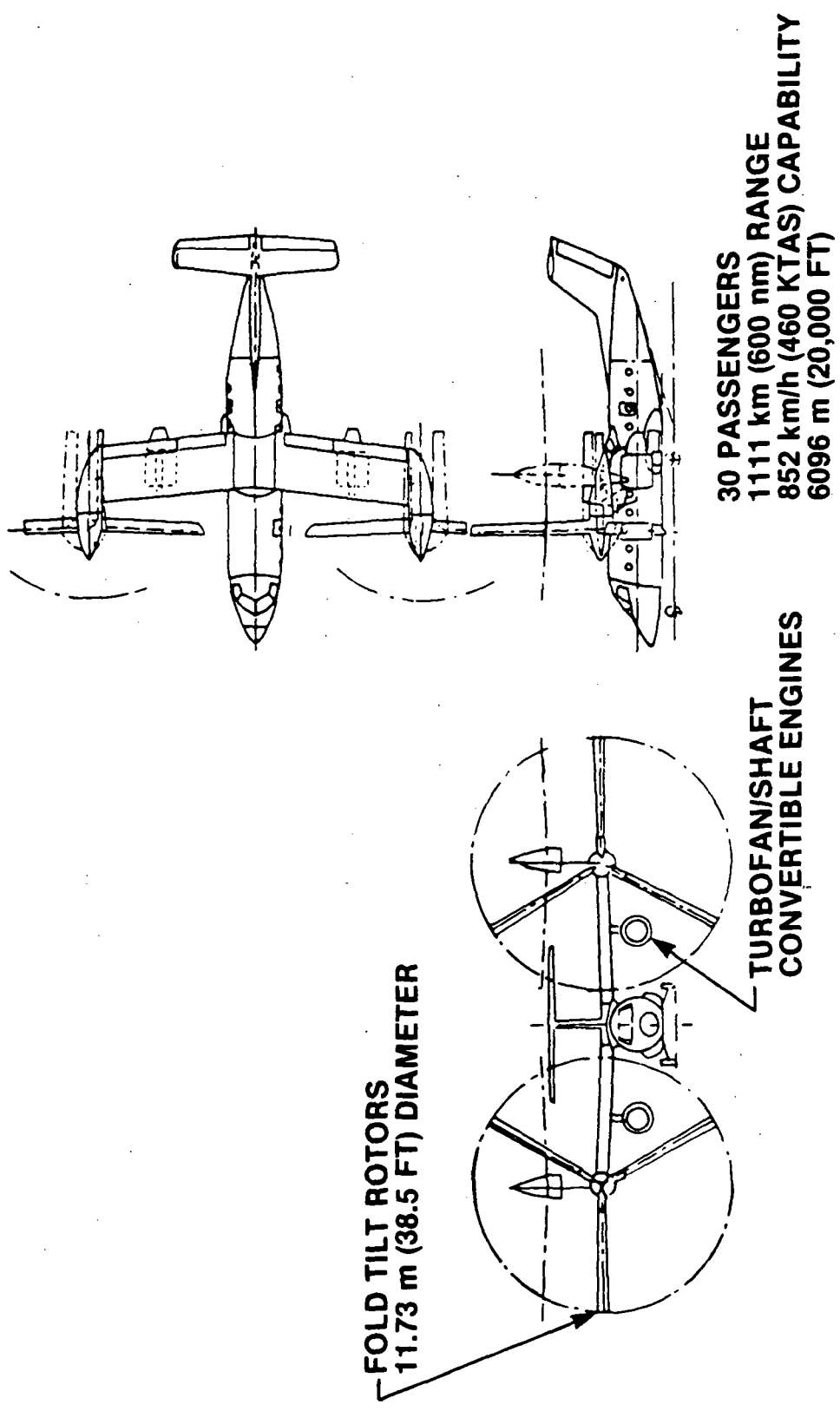
### III. AIRCRAFT CONSTRAINTS

The torque converter design requirements chosen were for the fold tilt rotor application and were based on the commuter transport shown in Figure 1 (reference "Rotorcraft Convertible Engine Study," Contract NAS3-22742, NASA CR 168161). The aircraft was designed to carry 30 passengers with an unrefueled range of 1111 km (600 nm) and a maximum speed capability of 852 km/h (460 KTAS) at 6096 m (20,000 ft). The three-view sketch shows the 11.73 m (38.5 ft) diameter proprotors in the airplane flight mode with the prop mast tilted to the forward flight position and the proprotors shown in both the extended and folded position. The sketch also shows the wing tip pods tilted upward with the proprotor mast vertical for flight in the helicopter mode.

The convertible engines are located in nacelles below the wing. The engines are cross-shafted so that both engines or either engine alone can drive the proprotors. A device for decoupling the rotor drive shaft when the rotors are stopped is included in the aircraft power transmission system.

The fold tilt rotor aircraft is a variant of the XV-15 tilt rotor aircraft but differs in that the proprotors can be stopped in flight, feathered, indexed, and folded. Propulsion of the aircraft is transferred to the turbofan mode of operation, which permits efficient operation at higher aircraft speeds. Thus, the fold tilt rotor aircraft permits further expansion of the aircraft operating envelope over that of the XV-15 and conventional helicopter and fixed wing aircraft, as shown in Figure 2. With the rotors folded and thrust supplied by turbofan engines, the convertible engine powered aircraft flight envelope is expanded to 0.75 Mach. With the rotors deployed, a flight envelope similar to that of the XV-15 could be obtained in the helicopter and turboprop modes.

The folding proprotor principle has been demonstrated in wind tunnel tests. A 1.5 m (5 ft) model was successfully run in 1971, and a 7.6 m (25 ft) folding proprotor (the same diameter used in the XV-15) successfully completed testing in 1972.



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Figure 1. Fold tilt rotor commuter aircraft with convertible engine.

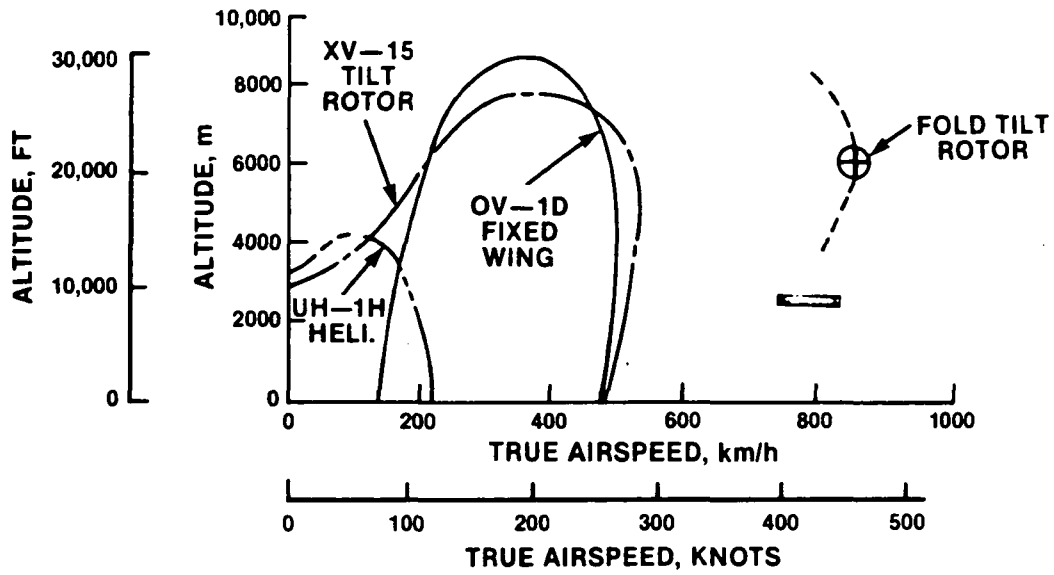


Figure 2. Flight envelope comparisons.

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#### IV. TORQUE CONVERTER CONSTRAINTS

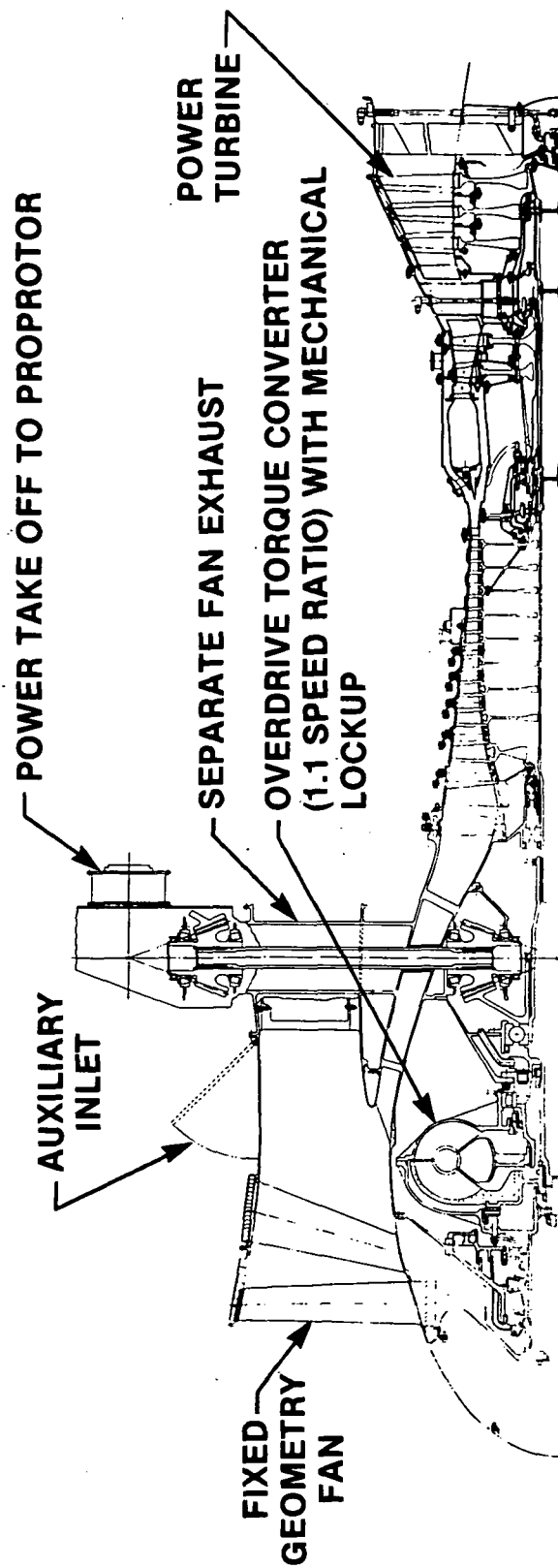
The convertible engine arrangement is shown in Figure 3. The engine is a conventional turbofan engine with a fixed geometry fan driven by a two-stage power turbine. In the fan mode, the engine delivers the cruise performance shown in Table I. An overdrive torque converter with a mechanical lockup is located in the engine inlet housing and serves to decouple the fan from the power turbine shaft. Shaft power is provided to the aircraft through the power takeoff drive to drive the proprotors in the helicopter and turboprop flight modes. In the turboshaft mode, the engine provides the shaft power shown in Table I. Auxiliary inlets are open in the turboshaft mode to provide additional flow area for air entering the gas generator compressor.

Table I.  
The convertible turbofan/turboshaft engine power requirements

Turboshaft mode sea level, 32.2°K (90°F) max		Turbofan Mode 0.75 Mach, 6096 m (20,000 ft) max continuous	
Power--kW (shp)	12,389 (6275)	Thrust, $F_n$ --N (lbf)	13,500 (3035)
SFC--mg/W-h (lbm/shp-hr)	239.9 (0.394)	TSFC--mg/N-s (lbm/lbf-hr)	18.44 (0.651)
Pressure ratio, $R_c$	24.5	Bypass ratio	6.85
RIT--°K (°F)	1700 (2600)	Pressure ratio, $R_c$	30.2
Power turbine speed, $N_{PT}$	100%	RIT--°K (°F)	1589 (2400)

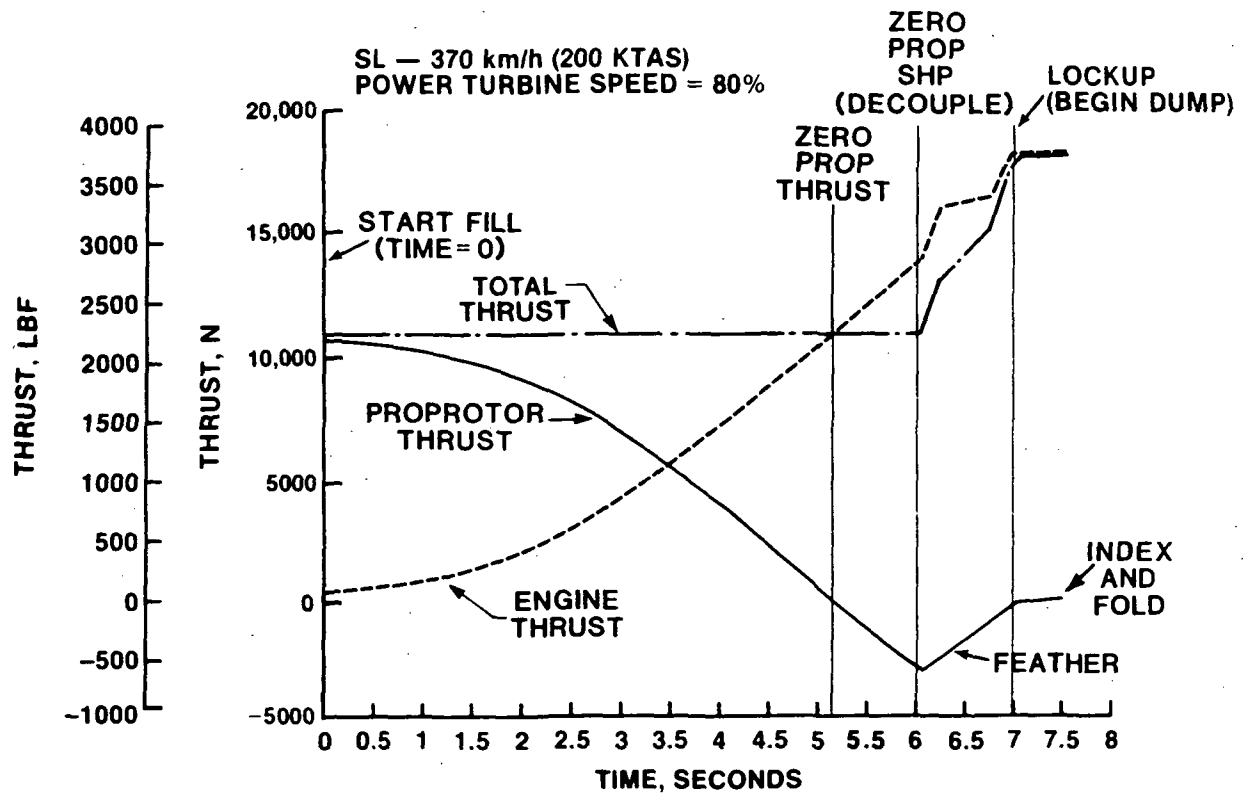
The torque converter is normally empty but is filled during fan coupling to accelerate the fan to input/output shaft synchronous speed to permit engagement of the locking device. It is then emptied to eliminate oil churning losses. The torque converter is refilled to disengage the mechanical lockup and emptied to decouple the fan.

The time history of thrust during the turbofan-to-turboshaft conversion is shown in Figure 4 as generated by a computer model. The conversion is shown for a flight condition of 370 km (200 KTAS) at sea level. Power turbine speed is held constant at 80% during the process. A total thrust of 9919 N (2230 lb) is required per engine to fly the aircraft at the condition selected. Total thrust is held constant until the proprotor is decoupled and the turbofan takes over the propulsion function. The conversion is initiated at time zero when oil starts filling the torque converter. As the fan accelerates and absorbs more power, the proprotor blades change pitch to absorb less power and hold speed constant at 80%. The prop reaches a zero thrust condition and begins windmilling. At the zero torque condition, the proprotor is decoupled by a clutch in the aircraft drive system. Power turbine speed is then held constant by the engine power turbine governor. The proprotor is feathered, indexed, and folded while the fan continues to accelerate to synchronous speed and the mechanical lockup in the torque converter is engaged. The torque converter is then emptied of oil. Fan shaft and power turbine lockup is obtained in approximately 7 sec.



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Figure 3. Convertible turbofan/turboshaft engine.



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Figure 4. Computer model--turboshaft-to-turbofan conversion.



## V. TORQUE CONVERTER REQUIREMENTS

From these preliminary studies, the torque converter design requirements were established as follows:

- o application to a turbofan/shaft engine in the 4846 kW (6500 shp) class
- o flightweight
- o internally housed
- o torque converter mode
  - o 5077 N-m (3744 lbf-ft) torque at 6986 rpm 3743 kW (5020 shp)
  - o mechanical lockup at 6986 rpm (80%)
  - o output power (after lockup) 5741 kW (7700 shp)
  - o acceleration time goal 5-10 sec
- o lockup mode rotor speeds
  - o aerodynamic mechanical design ( $N_D$ )--8734 rpm (100%)
  - o maximum steady state ( $N_S$ )--9083 rpm (104%)
  - o maximum transient ( $N_T$ )--9345 rpm (107%)
  - o rotor integrity ( $N_I$ )--10,446 rpm (119.6%)
  - o design burst ( $N_B$ )--11,083 rpm (126.9%)
  - o minimum idle speed--5677 rpm (65%)
- o life goals
  - o military--5000 hr
  - o commercial--25,000 hr
- o low cycle fatigue goals
  - o military--12,000 cycles
  - o commercial--50,000 cycles

## VI. CONCEPTUAL DESIGN

### GENERAL ARRANGEMENT

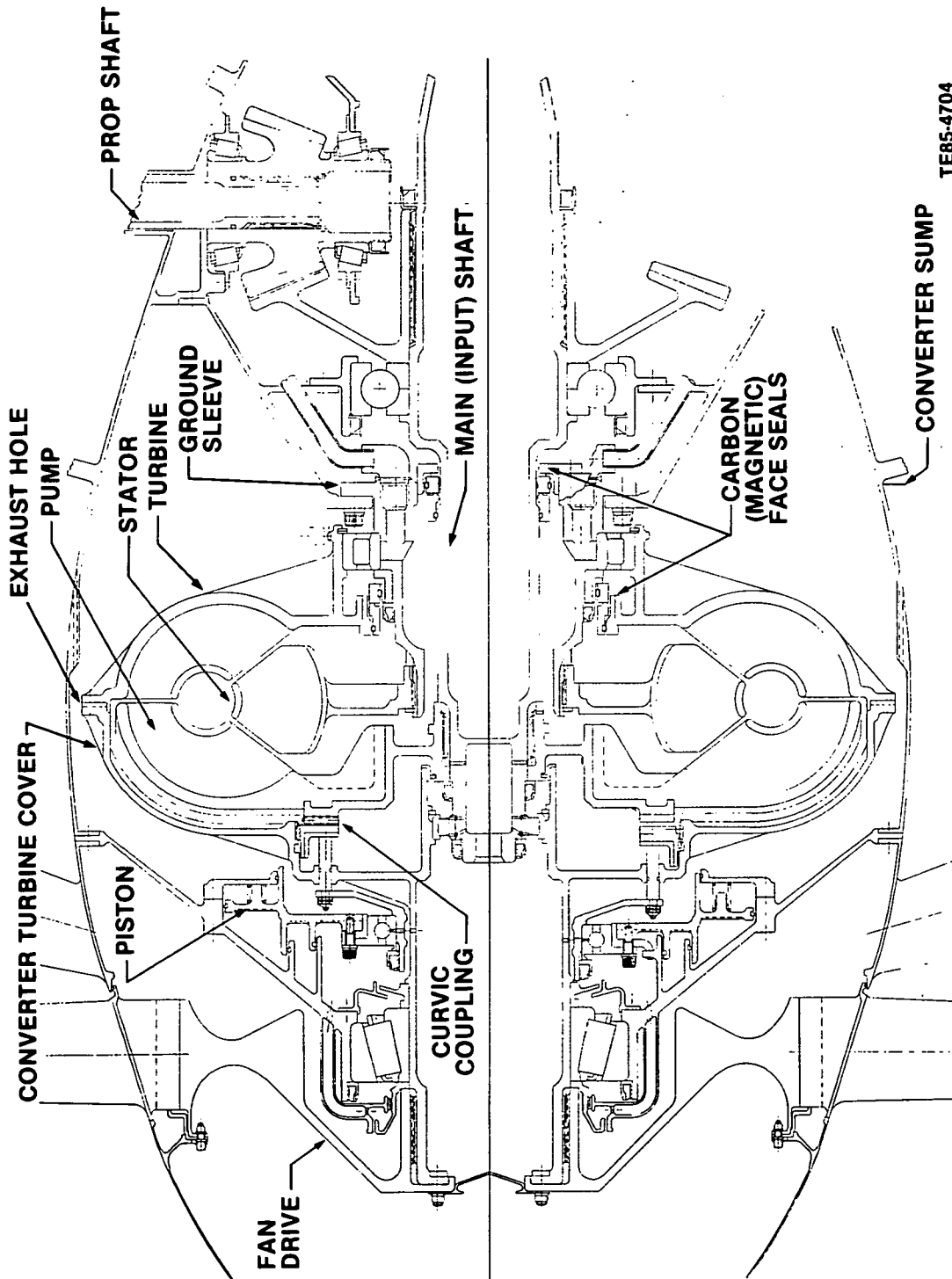
The convertible engine torque converter concept consists of two major systems, the torque converter and the mechanical coupling. The torque converter is a hydrodynamic unit used to transfer power from the low pressure (LP) turbine of the engine to the fan during transition from prop rotor mode to fan mode and vice versa. A detailed description of the operation and performance of this unit is provided in the performance and operation subsection. The second major system is the mechanical coupling. This unit transfers power from the engine to the fan during cruise operation. A discussion of this device can be found in the subsection entitled Mechanical Coupling in Section III. A cross section sketch of the convertible engine torque converter is shown in Figure 5. The torque converter and mechanical coupling are housed in the front section of the engine. The walls in this area form a sealed internal cavity for these components. A converter sump is incorporated into the lower section of the cavity. Oil from the mechanical coupling and torque converter collects in this area before being pumped back into the engine oil system.

#### Torque Converter

The torque converter is made up of three elements: pump, turbine, and stator. The pump is coupled to the engine main shaft via the pump hub and runs at low pressure turbine speed. A fixed spline is used to couple the main shaft to the pump hub. This spline is lubricated by engine oil supplied through the middle of the main shaft. The pump hub is made of steel to accommodate the curvic coupling and main shaft spline. The outside of the hub is splined or grooved, and the titanium for the pump section is cast on the grooved hub. This is a technique recently introduced to reduce machining time and weight. Weight is reduced by the elimination of bolts required to hold the two pieces together. A roller bearing at the left end of the shaft and a ball bearing at the right end (around the prop drive bevel gear set) provide support for the pump section. The converter turbine is located to the right of the pump section. Fluid motion between the pump and turbine blades causes the turbine to rotate. Turbine rotation is transferred to the fan via a turbine cover that is bolted to the turbine. A tapered bearing at the front of the engine next to the fan and a roller bearing below the turbine provide support for the turbine cover assembly. The main purpose of the tapered bearing, however, is to support fan and converter turbine thrust loads.

The stator directs oil from the turbine back to the pump. This unit is splined to a stationary ground sleeve. Unlike conventional torque converters where the stator is locked at low speed ratios and allowed to rotate (free wheel) at high speed ratios, the convertible engine torque converter stator is always stationary. The reason for this, as will be explained in more detail in the performance subsection, is that it allows the speed ratio between the pump and turbine to reach and even go above 1:1. The 1:1 speed ratio is important in this concept since it is the point at which the mechanical coupling is engaged.

Three rotating seals are employed to keep oil in the converter cavity. Pressure balanced carbon face seals are used in the two locations that experience both high relative speed and high oil pressure simultaneously. These locations



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Figure 5. Convertible engine torque converter cross section.

are (1) input shaft to ground sleeve and (2) turbine to ground sleeve. The magnetic face seals used in these locations are very compact, and the magnetic force between the two parts of the seals results in a very consistent and uniform force on the seal face. A rotating cast iron hook joint seal is used between the input shaft and the converter turbine cover/hub. This location does not experience both high pressure and high speed simultaneously. Instead, when the oil in the converter cavity reaches maximum pressure, there is very little relative speed between the input shaft (pump) and turbine cover (turbine).

### Mechanical Coupling

A curvic (face spline) coupling is used in the convertible engine torque converter concept to couple the engine low pressure turbine to the fan. One half of the coupling is splined to the main shaft via the converter pump hub. The other half is splined to the fan via the converter turbine cover/hub. This half of the coupling can be moved axially for engagement or disengagement by movement of the apply/disengage piston. The piston assembly is nestled into the front wall of the engine to minimize the length of the mechanical coupling assembly. This device is explained in more detail in a later subsection.

### HYDRAULIC SCHEMATIC

The convertible engine torque converter hydraulic system consists of two main sections. The first section uses a converter charging pump that supplies oil to the converter during transition from one operating mode to another. The other section uses engine main oil pressure to activate the pistons of the mechanical coupling and to lubricate the splines and bearing in the converter section. The combined system, shown in Figure 6, is an integral part of the engine oil system and uses a common oil tank.

### Converter Charging Circuit

During transition modes, the torque converter requires 3.8 L/s (60 gal/min) oil flow for filling and charging to full capacity. This is explained fully in the performance and operation section. A gerotor pump operating at 10,000 rpm is employed to supply the required flow at approximately 1379-1723.5 kPa (200-250 lb/in.<sup>2</sup>). Integral with this charging pump is a scavenge pump. Both pumps are driven by the gas generator through a single electric clutch that is engaged during converter operation. Therefore, these pumps function only during transition modes. The scavenge pump supplements the existing engine scavenge system during these modes. During normal operation, oil in the converter sump is removed by the engine scavenge system. The clutch capacity required to engage the charging and scavenge pumps is approximately 13.5 N-m (10 lbf-ft) at the operating speed of 10,000 rpm. Presently available commercial clutches in this torque range are limited to much lower speeds. A detailed design/development effort will be required to produce the clutch selected for this application. An alternate scheme, using a 5000-rpm pump and clutch (see the subsection entitled Alternate Schemes), can be used in a demonstration program where a flightweight unit is not required.

The charging circuit uses one relatively simple valve, a two-stage solenoid regulating valve. The first stage regulates pressure between 0-345 kPa (0-50 lb/in.<sup>2</sup> gage) while the second stage regulates between 345 and 2070 kPa (50

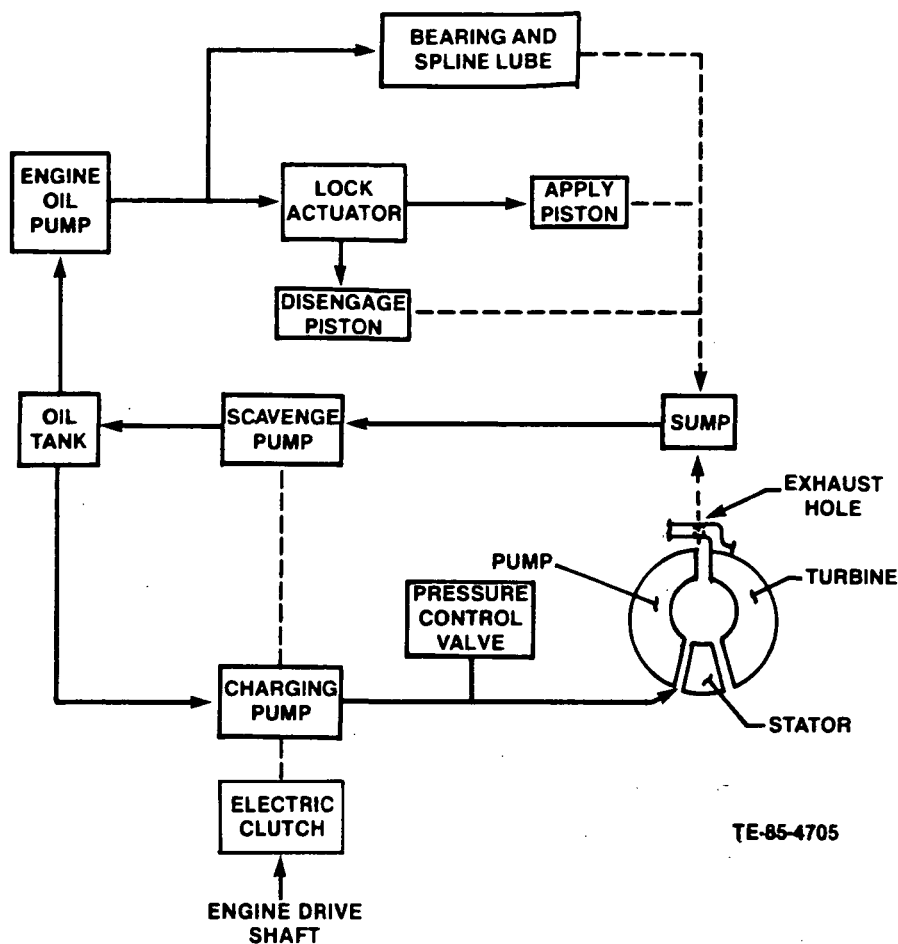


Figure 6. Hydraulic schematic.

and 300 lb/in.<sup>2</sup> gage). Converter oil flow will not pass through the valve. Instead, the valve will allow charge pump flow to bypass to sump as required to regulate converter inlet pressure. Oil is exhausted from the converter through fixed orifices at the outer diameter of the converter. Flow through these holes provides cooling during converter operation in the transition mode and acts as the converter dump valve to evacuate the oil in the converter after the mechanical coupling is engaged and the converter charging pump is decoupled.

#### Pistons and Lube Circuit

This section of the hydraulic circuit is less defined at this point of the design. Since it is an integral part of the engine main oil circuit, further design of the system will be done in conjunction with the design of the engine oil system. In concept, part of the engine main oil flow is routed to a lockup control valve. This valve routes the oil to the apply or disengage piston as required. Oil from the engine main circuit is also used to lubricate the splines and bearings in the converter section.

## OIL PASSAGES AND POWER FLOW

### Oil Passages

Oil from the converter charge pump is fed into the torque converter through the ground sleeve. To provide uniform inlet flow, four inlets are equally spaced. Inlet flow goes from the ground sleeve along the gap between the ground sleeve and main shaft and then into the converter pump. While the converter cavity is being filled, oil leaves the cavity at the outer diameter of the converter shell. Four holes are on the converter to provide uniform outlet flow. These holes are sized so that the outlet flow rate is less than the inlet flow rate, thus making it possible to fill the converter and charge it to the pressure required for synchronous speed between the pump and turbine. Once the inlet flow is cutoff, these four holes act as a drain for the converter. The fill-dump cycle will be explained in full detail in the performance subsection.

Oil from the engine main pressure circuit is fed to the apply and disengage pistons through lines in the front wall of the compartment. Lube oil for all the bearings (except converter turbine roller bearing) and splines in the cross section (Figure 7) is supplied through the middle of the main shaft. Centrifugal force is then used to distribute the oil radially to the bearings and splines.

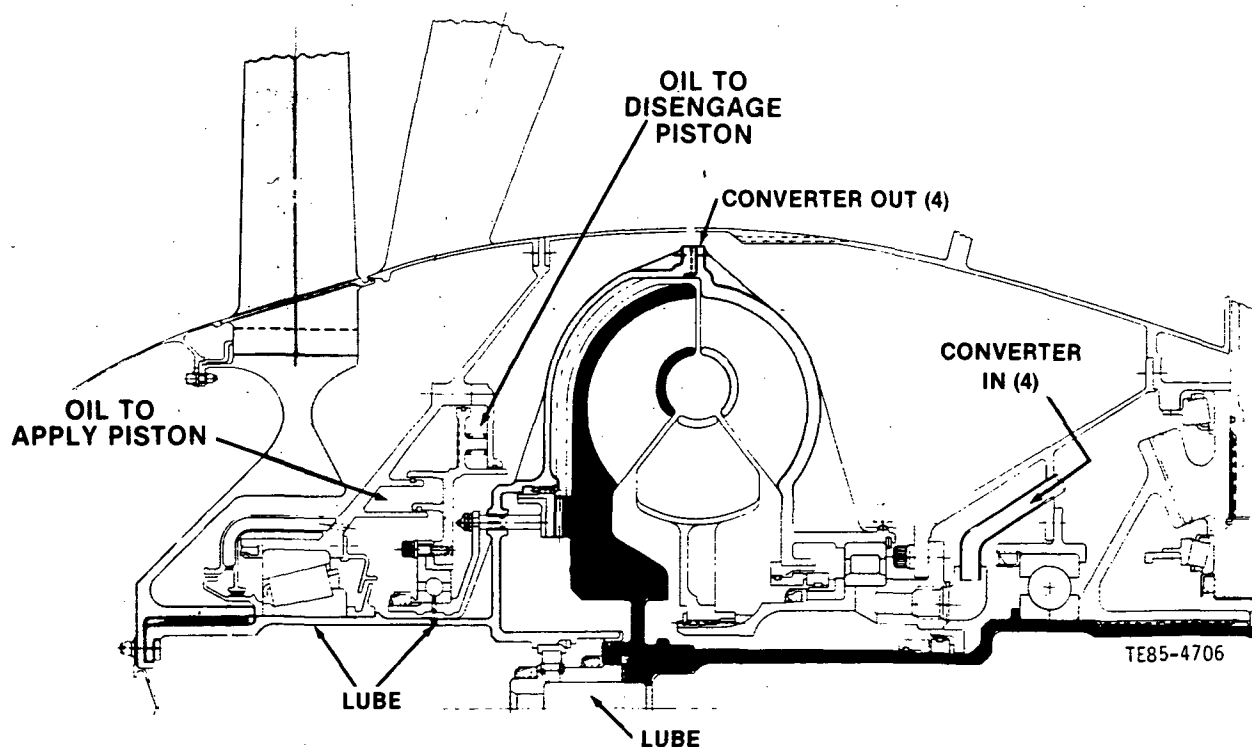


Figure 7. Oil passages.

### Power Flow

During the prop rotor operation, the torque converter is empty and engine power goes entirely to the propeller. As oil enters the converter some of engine power is diverted to the fan. When the converter reaches maximum capacity, all the engine power (minus losses) is delivered to the fan via the torque converter. This power transfer from engine to fan is achieved by transfer of energy across the fluid link between the converter pump and turbine. After synchronous speed is achieved between the pump and turbine, the mechanical coupling is engaged and the oil feed to the torque converter is cut off. Once the converter cavity is drained, all the engine power is transmitted to the fan through the mechanical coupling. These power flow paths are shown in Figures 8 and 9. During the process of power transfer to the fan, the prop rotor is decoupled at zero torque, stopped, indexed, and folded to a stowed position for flight in the turbofan mode.

### MECHANICAL COUPLING

The mechanical coupling scheme designed to transmit power from the engine to the fan during fan thrust mode is shown in Figure 10. The scheme consists of a stationary, one-piece apply and disengage piston, a ball bearing employed to transfer the axial motion of the piston to a rotating apply arm, and a curvic (face spline) coupling. Conventional torque converter lockup clutches (mechanical couplings) employ rotating clutches and pistons. This requires the

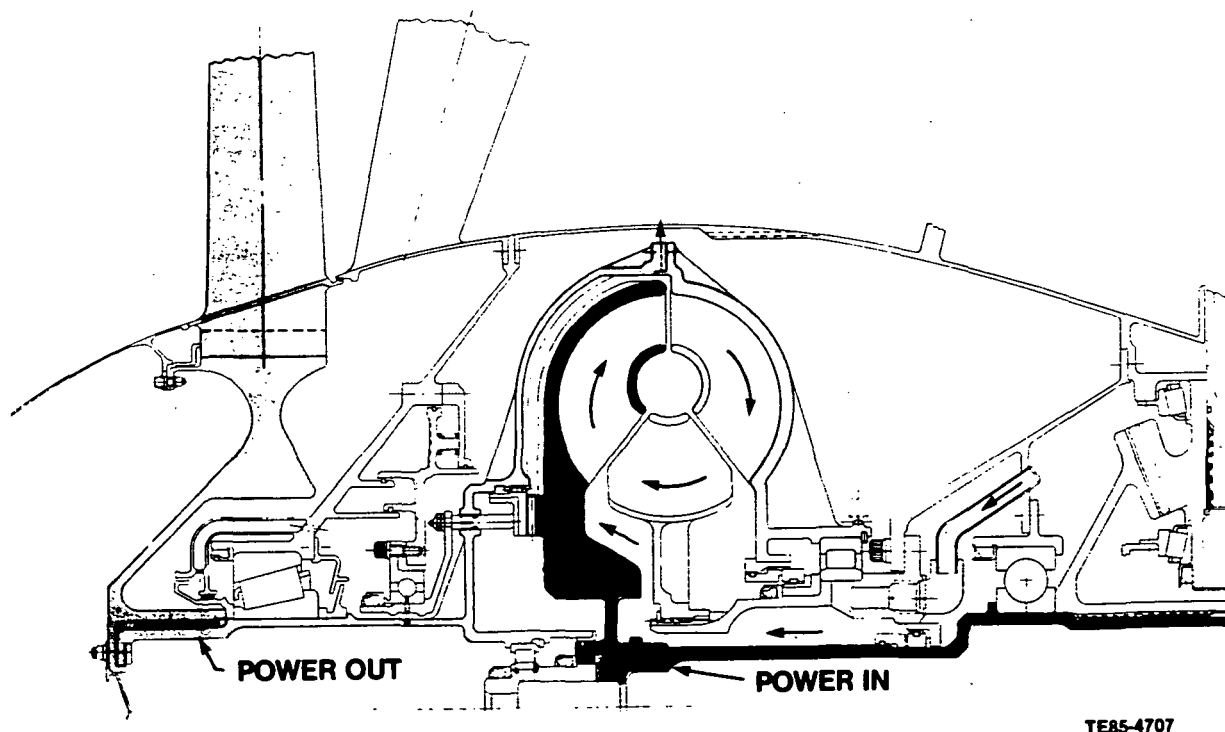


Figure 8. Power flow--fluid.

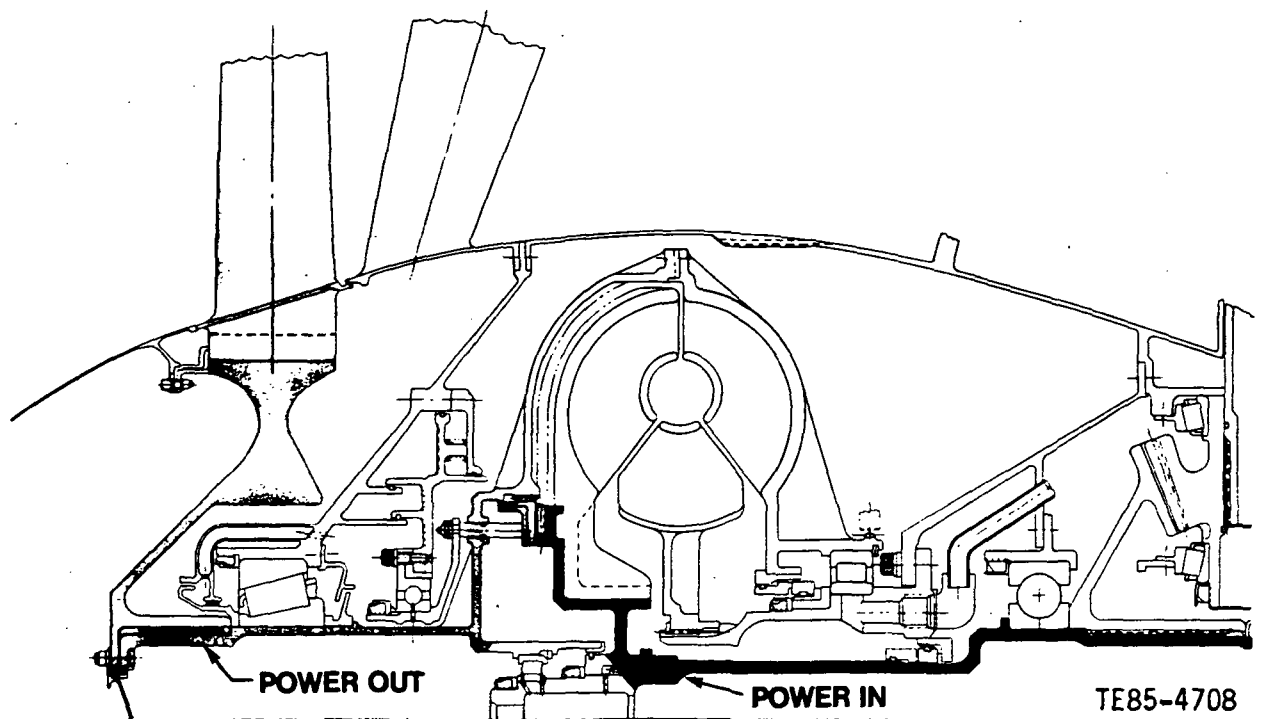


Figure 9. Power flow--mechanical.

- **STATIONARY PISTONS**
  - NO ROTATING SEALS
- **ALUMINUM PISTON AND PISTON HOUSING**
- **CURVIC (FACE) COUPLING**
  - DESIGNED FOR ENGAGEMENT UNDER LOAD
  - EVERY OTHER TOOTH MISSING AND TOOTH TIP ROUNDED FOR EASE OF ENGAGEMENT
  - NEGATIVE ANGLE KEEPS COUPLING ENGAGED IF APPLY PRESSURE IS LOST
  - DISENGAGE PISTON SIZED TO DISENGAGE COUPLING UNDER PART LOAD

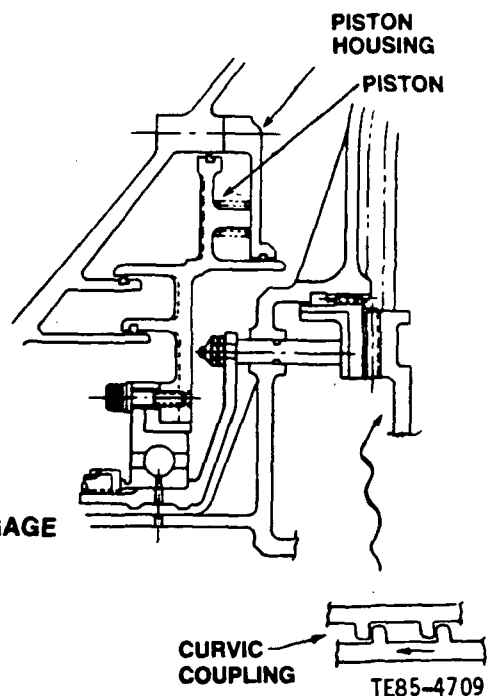


Figure 10. Mechanical coupling.



transfer of pressurized clutch apply oil across rotating seals. This becomes a problem at very high pressures and/or rotating speeds. These conditions can result in PV valves exceeding the capability of most seals. To avoid this problem, the convertible engine torque converter mechanical coupling is designed with a nonrotating piston. This greatly simplifies the oil feed to the piston since lines can be routed directly to the piston cavity.

The piston housing is machined into the engine front wall and the disengage piston bore is stacked radially outward of the apply piston to minimize the overall width of the piston assembly. The one-piece piston is coupled to the outer race of a ball bearing. The inner race of the bearing is coupled to the part of the curvic coupling rotating with the turbine cover via an apply arm. Axial movement of the piston causes the bearing to move axially. The rotating inner race transfers this motion to the curvic coupling. By employing a double acting piston, the coupling can either be moved into or out of engagement. The piston is designed to operate with engine main oil pressure equal to or greater than 276 kPa (40 lb/in.<sup>2</sup>). Pressurizing the apply piston cavity moves the piston to the right and engages the coupling, whereas pressurizing the disengage piston cavity moves the piston to the left and disengages the coupling.

A curvic coupling (face spline) is employed to transmit torque from the pump hub (engine) to the turbine cover (fan). To facilitate engagement of the coupling, every other tooth is removed. This provides a relatively large gap between teeth, which allows easy engagement. Engagement is also enhanced by rounding off the top of each tooth as shown in Figure 10. The face of the tooth is machined at a negative 5 deg angle. This feature keeps the coupling engaged if apply piston pressure is lost during flight. However, the negative angle coupling requires more force for disengagement while transmitting torque. As a result, the disengage piston is larger than the apply piston. The disengage piston is designed with sufficient capacity to disengage the coupling when it is transmitting up to 50% of maximum torque--such operation may be desired if the converter fails to perform at maximum capacity during transition from fan to propeller mode.

#### CONTROL LOGIC

The convertible engine torque converter controls will be an integral part of the engine control system. The converter control logic shown in Figure 11 is made up of the following parts:

- o input speed sensor--monopole magnetic pickup
- o output speed sensor--monopole magnetic pickup
- o lock position sensor--monopole magnetic pickup
- o lock pressure sensor--pressure switch
- o charging pressure sensor--integrated circuit strain gage sensor 0-2068 kPa (0-300 lb/in.<sup>2</sup> gage)
- o lock/unlock valve--four way valve
- o pressure control valve--0-2068 kPa (0-300 lb/in.<sup>2</sup> gage) two-stage high flow valve

Control of conversion to fan thrust mode will require communication between the integrated engine/converter control system and the aircraft flight controllers. The integrated control can take on the responsibility of providing

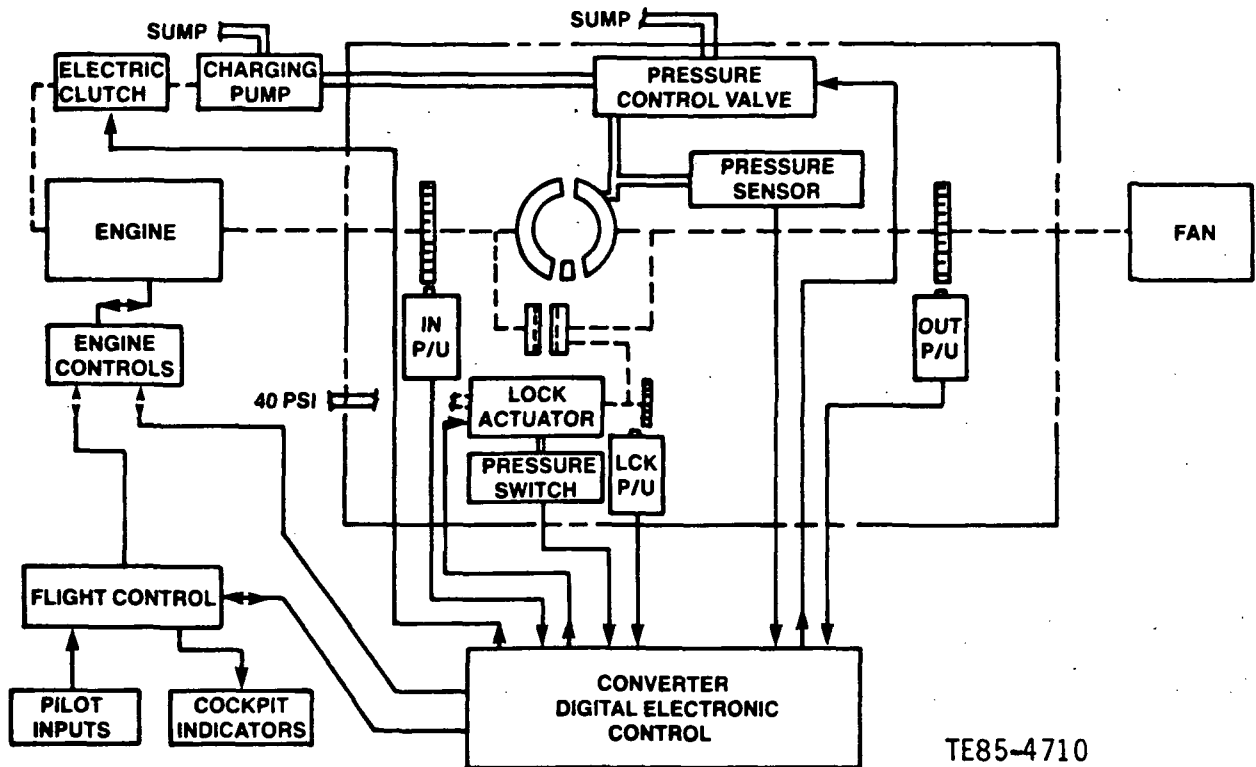


Figure 11. Control logic.

a smooth transition of power to the fan section of the engine while the propellers are being unloaded. When the converter has achieved a synchronized condition of input and output speeds, the control will automatically lock up the shafting and disengage the converter charge pump, thus cutting off oil flow to the converter. This allows the converter to drain for maximum efficiency operation in the turbofan operating mode. Conversely, to make a transition from the fan thrust mode to prop rotor mode, the control will unload the lock mechanism by filling the converter, command and monitor the lock retraction, then program unloading of the fan while the propellers are being accelerated.

## MATERIAL CONSIDERATIONS

Most conventional torque converters for automotive application are made of cast aluminum or stamped steel. The size and speed of these converters result in a blade tip speed of approximately 73 m/s (240 ft/sec). The large diameter and high speed of the convertible engine torque converter results in a blade tip speed of 171 m/s (560 ft/sec). Also, this converter is subjected to much higher internal oil pressures. These conditions give rise to relatively high stresses in the converter sections. This led to the examination of several materials for use in these sections. The different materials were evaluated on the basis of cost, strength, and weight. The results are summarized in Table II. Titanium was chosen for the converter sections due to its low weight and high strength. The piston and piston housings are made of aluminum for low weight while the shafts, hubs, and couplings are made of steel for high strength.

Table II.  
Convertible engine torque converter material considerations.

<u>Material</u>	<u>Cost</u>	<u>Strength</u>	<u>Weight</u>
Steel	Low	Very high	Very high
Titanium	Moderate	High	Moderate
Aluminum	Low	Very low	Low
Composites	High	Moderate	Very low

- o Mechanical coupling
  - o aluminum piston and housing
  - o steel arm, splines, and hub
- o Torque converter
  - o investment cast Ti 6-4 pump, turbine, and stator
  - o steel input and output shafts

#### MANUFACTURING PROCESS

Having chosen titanium (Ti 6-4) as the material for the convertible engine torque converter pump, turbine, and stator sections, several manufacturing processes were investigated. Five manufacturing processes are available: forging, investment casting, sand casting, die casting, and sheet metal fabrication. In automotive applications where rotating speeds and loads are relatively low, sand or die cast aluminum and sheet steel fabrication are used. In aircraft applications where a high strength-to-weight ratio is required, titanium forging and investment casting processes are used to manufacture those components that are similar in shape to the torque converter.

Table III gives a qualitative summary of the five manufacturing processes in four categories: unit cost, tooling cost, performance, and strength. In each category, a scale from 1 to 10 is given where 1 and 10 represent the worst and best process, respectively. A process with the highest total number is judged to be the best process. It can be seen from Table III that investment casting is chosen to be the best process to manufacture the convertible engine torque converter. Titanium forging/machining, a process that should yield very high strength, is rated second. This is a result of the complex shape of the torque converter sections. Because of this complexity, the inner core, blades, and outer shell cannot be machined simultaneously in one piece. The blades and shell surface must first be machined and then the inner core surface must be mounted to the blades by brazing, welding, or riveting. The moderate score results not from the forging process but from the mounting (braze/weld/rivet) process. If a high quality of weld can be achieved, the forging process can be very competitive for the manufacture of a convertible engine torque converter. The other three processes are not recommended for the following reasons:

- o Titanium material cannot be cast with the sand and die casting processes. A controlled environment is needed to cast titanium material.
- o Sheet metal stamping cannot provide enough strength and performance for the convertible engine application.

Table III.  
Manufacturing process trade-off with titanium material.

<u>Process</u>	<u>Unit cost</u>	<u>Tooling cost</u>	<u>Performance</u>	<u>Strength</u>	<u>Total</u>	<u>Rank</u>
Sand casting	5	8	1	1	15	3rd
Die casting	3	5	1	1	10	5th
Investment casting	3	3	9	10	24	1st
Forging (machining)	1	5	10	5	21	2nd
Sheet metal stamping	10	1	1	1	13	4th

Specialized technology is required to produce a high quality titanium investment casting. However, three vendors in the United States indicated that they had the technology required to produce castings for this project.

#### VIBRATION CONSIDERATIONS

A detailed vibration analysis was not performed on the convertible engine torque converter due to the limited scope of this program. Also, the uncertainty of exactly how the system will be integrated with the engine makes it difficult to perform a detailed analysis. However, the system was evaluated qualitatively to ensure that it is structured such that it can be analyzed and modified during a design phase to move any natural frequencies outside of the operating range. Also, steps were taken to ensure even flow distribution into and out of the torque converter to minimize vibrations during the dump and fill cycles. Finally, a different number of blades were used in the pump, turbine, and stator to reduce vibration during converter mode.

#### STRESS SUMMARY/LIFE FATIGUE

The design criteria for the convertible engine torque converter are as follows:

- o fan/shaft convertible engine application, 4846 kW (6500 shp) class
- o lightweight
- o internally housed
- o torque converter mode
  - o 6986 rpm, 5077 N-m (3744 lbf-ft), 3743 kW (5020 shp)
  - o mechanical lockup at 6986 rpm (80%)
  - o output power (after lockup) 5742 kW (7700 shp)
  - o acceleration time goal: 5-10 sec
- o lockup mode rotor speeds
  - o aerodynamic mechanical design ( $N_D$ )--8734 rpm (100%)
  - o max steady state ( $N_S$ )--9083 rpm (104%)
  - o max transient ( $N_T$ )--9345 rpm (107%)
  - o rotor integrity ( $N_I$ )--10,446 rpm (119.6%)
  - o design burst ( $N_B$ )--11,083 rpm (126.9%)
  - o minimum idle speed--5677 rpm (65%)
- o life goals--military 5000 hr, commercial 25,000 hr
- o low cycle fatigue goals--military 12,000 cycles, commercial 50,000 cycles

Using the conditions outlined in the torque converter mode and lockup mode, critical areas of the torque converter/mechanical coupling scheme were analyzed and the results are listed in Table IV. The analysis performed was brief but sufficient to determine the feasibility of the concept. A more detailed finite element analysis will have to be performed during the design phase to optimize the weight of the unit. Of all the locations analyzed, the stresses were not high enough in any area to produce low cycle fatigue. Based on this study, the convertible engine torque converter should meet the goal of 50,000 commercial cycles.

Table IV.  
Stress summary.

<u>Location</u>	<u>Basic stress--kPa (lb/in.<sup>2</sup>)</u>	<u>Surface stress--kPa (lb/in.<sup>2</sup>)</u>	<u>Allowable stress--kPa (lb/in.<sup>2</sup>)</u>
Curvic coupling (face)		39,714 (5,760)	1,034,213 (150,000 [shift under load--case hard])
Spline (curvic cpl O.D.)		30,668 (4,448)	82,737 (12,000 [working])
Spline (input shaft)		100,808 (14,621)	103-172,369 (15-25,000 [fixed])
Shaft (input)	284,436 (41,254)		275-413,685 (40-60,000)
Hub (conventional pump)	91,342 (13,248)		275-413,685 (40-60,000)
Shaft (output)	113,784 (16,503)		275-413,685 (40-60,000)
Shell O.D. (at 119.6% speed)	385,617 (55,929)		613,633 (89,000)

#### WEIGHT ANALYSIS

A weight summary for the torque converter/mechanical coupling scheme is shown in Table V. Use of titanium and aluminum wherever possible accounted for the low weight of the complete system. The oil estimate in Table V is the amount of oil that will have to be added to the engine system to accommodate the torque converter.

Table V.  
Weight summary.

	<u>Weight--kg (lb)</u>	
Converter section (pump, turbine, stator)	51.3	(113)
Mechanical coupling (piston, housing, curvic)	16.3	(36)
Oil supply system (pump, clutch, lines)	3.2	(7)
Oil (~15.1 L [4 gal])	13.2	(29)
Controls (pressure regulator, speed pickup, etc)	4.5	(10)
Total	88.5	(195)

#### ALTERNATE SCHEMES

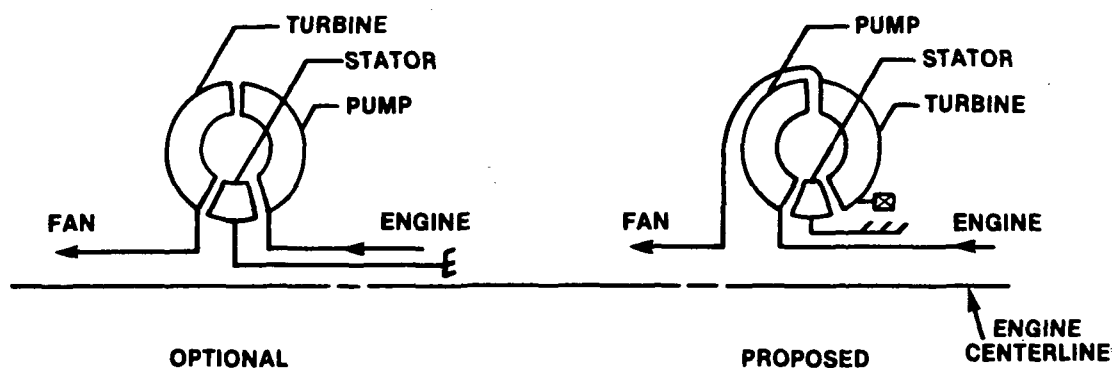
The system described in the previous subsections resulted from a lengthy feasibility study. During this study, several schemes were evaluated for the mechanical coupling, torque converter, and converter charge pump circuit. The

mechanical coupling schemes studied were a plate clutch, a sprag clutch, a helical spline coupling, and a curvic coupling. Two converter schemes were evaluated--the first had the turbine toward the front of the engine while the second had the pump toward the front of the engine (Figure 12). The charging pump schemes included a 10,000-rpm charging pump with a bypass valve, a 5,000-rpm pump with a 5,000-rpm electric disconnect clutch, and a 10,000-rpm pump with a 10,000-rpm electric disconnect clutch.

The plate clutch is used on conventional torque converters. These clutches require positive pressure to keep them engaged. Also, pressurized oil has to be transferred across rotating seals to activate the clutch piston. These two factors resulted in this option being eliminated. The sprag clutch and helical spline options with nonrotating pistons avoided these problems. However, both the sprag and the helical options require an indexing device to align the pump and turbine before the coupling is engaged. The curvic coupling described previously eliminated all the above problems and was therefore selected for this concept.

The first converter scheme studied had the converter pump close to the low pressure turbine and the converter turbine toward the fan. To ground the stator, a shaft had to be run through the center of the main shaft to the back of the engine. This was not a feasible option. Therefore, the converter was reconfigured as shown in Figure 5, with the stator grounded to the engine housing via the ground sleeve.

The 10,000-rpm converter charging pump and electric disconnect clutch were selected for the convertible engine torque converter concept to optimize the size and weight of the system. However, for a demonstration program where a flight-weight unit is not required, the 5000-rpm pump and clutch will be employed since they are readily available. The continuously flowing charging pump coupled with a bypass valve was eliminated for two reasons. First, the life of the pump is shortened since it works continuously. Second, the relatively large capacity of the converter charging pump will require extensive modification of the engine's air separator system to remove the air that will be added to the oil circuit if this pump runs continuously.



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Figure 12. Converter configurations.

## VII. PERFORMANCE AND OPERATION

### INTRODUCTION

There are two main potential advantages of using a torque converter for the application of fan/shaft convertible engines. First, the use of a torque converter will minimize the space requirement that allows it to be housed internally within the normal flow path lines of a turbofan engine. Second, the time required for a complete transition from prop rotor to fan mode (and vice versa) is very reasonable. The use of a torque converter permits decoupling the fan when not in use and operating the engine in the turboshaft mode. To convert from turboshaft to turbofan mode, the torque converter provides an overdrive feature to bring the fan speed up to the power turbine speed to permit engagement of a mechanical lockup device when the fan shaft and power turbine shaft speeds are synchronized. Thus, the conversion from shaft power can be made without droop of the power turbine speed that serves to minimize thrust loss during the transient, and the design of the mechanical lockup feature is simplified. This type of overdrive torque converter is known within General Motors Corporation, and experimental models have been built in sizes and power levels for use with normal automotive engines.

Underdrive torque converters with hydraulic clutches for mechanical lockup are common in automotive use. The application of an overdrive dump/fill torque converter at much higher power in an aircraft gas turbine engine requires that new technical information be generated that is applicable to the special requirements imposed. Torque converters have not been built and tested at the normal power and speed levels required for the convertible fan/shaft type of operation. The purpose of the torque converter conceptual design study is to identify these technical needs and to recommend a research and technology program to bring the concept to a state of technology readiness for full-scale development.

A baseline torque converter in automotive size was scaled to a size appropriate for the study. Within normal boundaries, power and speed as applied to power transmission in a torque converter vary as the cube of the speed. Converter test studies indicate that some deviation from a true cubic function occurs as the unit is operated at successively higher input power and speed. This deviation is mainly due to the effect of cavitation flow phenomenon that can be reduced by increasing the torque converter oil charging pressure and by obtaining a proper blade design configuration. This particular form of deviation or capacity drift occurs in all hydrodynamic devices, and these effects must be considered in determining geometry, blade shape and angles, diametral sizing, and charging pressure requirements. At a given speed ratio, the ability of a hydrodynamic torque converter to transmit torque varies as the square of the operating speed (i.e., the faster a converter is run, the more torque it will absorb or transmit until either hydraulic or mechanical limits are reached), and for each type and size of a converter, a specific capacity factor (C) is generated. The deviation noted in capacity at upper torque or power levels can be accounted for through proper diametral sizing, blade angles and shape, torus geometry, and converter charging pressure. All of the foregoing considerations must include weight and space claims in the chosen space frame. The scaled torque converter design general arrangement, features, and performance capabilities were estimated and test programs recommended to verify these estimates as appropriate.

The type of torque converter required for the convertible engine application has two features that differ from the torque converter used in the normal automotive application. First, in the automotive torque converter, one-way clutches are used to allow the stator element to resist torque in one direction, thus making it a reaction member and allowing it to rotate freely in the opposite direction when its function as a reaction member is not desired. For the application of the convertible engine torque converter where overdrive capability is required, one-way clutches must not be used. The stator element must be locked to a grounded member to provide underdrive and overdrive functions. Second, the general automotive torque converters are always operating under a fully filled condition. On the other hand, the convertible engine torque converter is operating as a dump/fill unit. Since the torque converter inlet charging pressure is basically zero during the partially filled conditions, serious cavitation problems can exist during transients for the convertible engine torque converter.

#### METHOD OF FILLING AND DUMPING

##### Filling mode

In the convertible engine fan drive application, the torque converter is empty at the beginning of the transition from shaft mode to turbofan mode or vice versa. When power transfer to the fan is initiated, the electric clutch connects the pump shaft to the engine, and the fluid is pumped into the torque converter circuit. The exhaust holes are located at the outer most radial locations as shown in Figure 5. Since the centrifugal effect causes the oil pressure to be maximum at these outer most radial locations, some volume flow rate will be exhausted through the exhaust holes. Along with the fan speed, the exhaust holes area and the radial positions of the exhaust holes determine the magnitude of the exhausted volume flow rate. Therefore, the difference between the pumped flow rate in and the exhausted flow rate out is the net flow rate retained in the torque converter circuit. The filling time is thus a function of the pumped flow rate in and the exhausted flow rate out.

Before the fully filled condition has been reached, the pumped flow rate in is always larger than the exhausted flow rate out; also, the torque converter inlet charging pressure is practically zero. When the torque converter is full, the pumped flow rate in is equal to the exhausted flow rate out, and the pressure regulating valve regulates the torque converter inlet charging pressure so that the torque converter has sufficient power transfer capability to achieve synchronous speed for mechanical lockup. After the mechanical lockup device has been engaged, the electric clutch disconnects the pump shaft from the compressor shaft, and the pump no longer pumps oil into the torque converter circuit. However, the oil can continue to flow out through the exhaust holes since the converter pump and turbine continue to rotate. Figure 13 shows a typical filling flow rate curve.

##### Dumping mode

During the lockup operation, the incidence and friction losses in the torque converter circuit result in some heat dissipation to the oil. This heat source is eliminated by dumping the oil from the torque converter and replacing it prior to transition back to the turboshaft mode. Oil flow into the torque



converter is stopped by use of an electric clutch that disconnects the pump shaft from the engine. The oil in the circuit is automatically dumped through the exhaust hole. Figure 13 shows a typical dumping curve.

## PERFORMANCE CHARACTERISTICS

The torque converter is a mixed-flow type of turbomachine that is used to transfer the power smoothly from the engine to the fan. It consists of a mixed-flow pump, a mixed-flow turbine, and an axial flow stator. The operating characteristics of the mixed-flow pump, mixed-flow turbine, and axial flow stator must be understood to design the three elements of the torque converter.

Improved understanding of these phenomena will significantly advance the hydrodynamic design technology of torque converter.

### Formulation for Scaling

Figure 14 shows a toroidal cross section of a typical torque converter. The size of a torque converter's element is characterized by the diameter, and the shape of the toroidal cross section can be expressed by a number of length ratios,  $l_1/D$ ,  $l_2/D$ ,  $l_3/D$ , and  $l_4/D$ . In addition to the geometric variables, as described above, the performance of a torque converter also depends on the control variables and the fluid properties. The control variables are the rotational speed of the element, the speed ratio between the output and input speeds, the torque converter inlet charging pressure, and the cooling flow rate. Density, viscosity, and specific heat are the three main fluid proper-

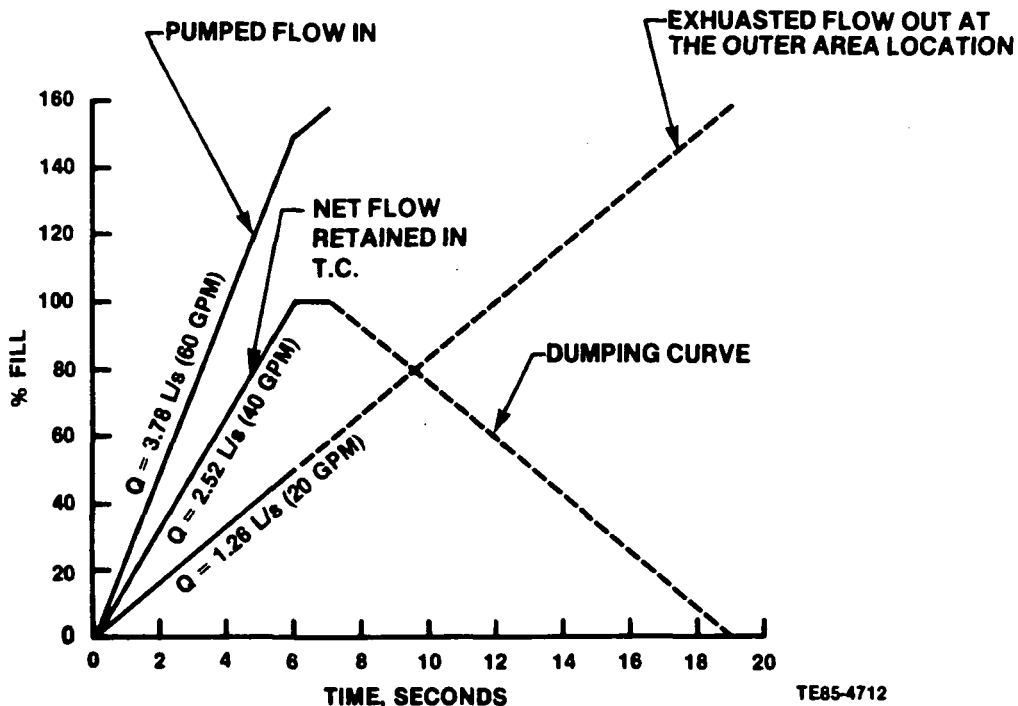


Figure 13. Method of filling and dumping for the convertible engine torque converter.

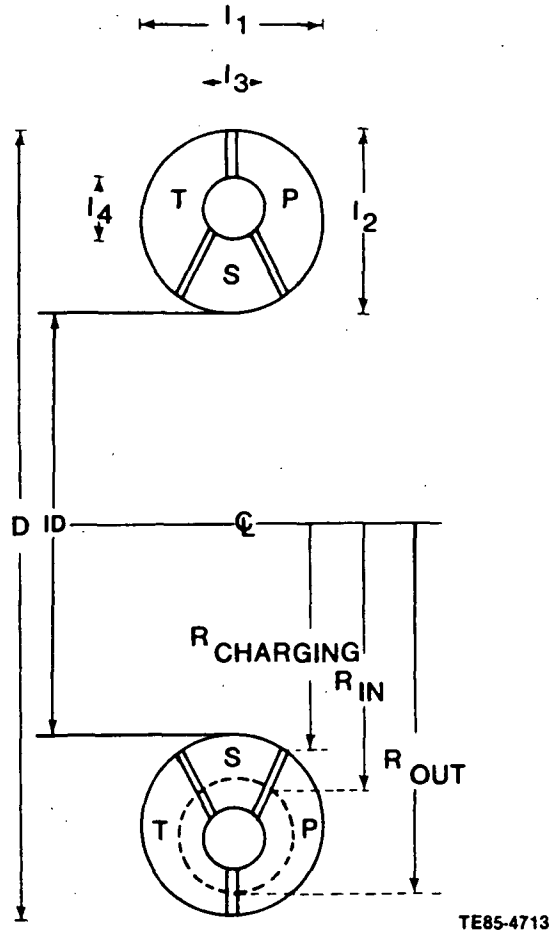


Figure 14. Toroidal cross section of a typical torque converter.

ties that can strongly affect the performance of a torque converter. For the torque converter input pump, the application of dimensional analysis leads to the following relationships:

$$\frac{\text{Power}}{\rho N^3 D^5} \text{ Pump} = P_{\text{pump}}, \text{ and,} \quad (1)$$

$$\frac{\text{Torque}}{\rho N^2 D^5} \text{ Pump} = T_{\text{pump}}, \text{ where,} \quad (2)$$

$\rho$	=	fluid density
$N$	=	rotational speed of the input pump
$P_{\text{pump}}$	=	power coefficient of the input pump
$T_{\text{pump}}$	=	torque coefficient of the input pump
$\text{Power}_{\text{pump}}$	=	power transfer through the pump element
$\text{Torque}_{\text{pump}}$	=	torque transfer through the pump element

In addition to the blade geometry (i.e., blade angles, blade spacings, blade profile, and blade thickness), the power and torque coefficients are functions of the length ratios, speed ratio, pressure ratio, Reynolds number, and surface roughness ratio. Mathematically, it can be written as follows:

$$P_{\text{pump}} = P_{\text{pump}} \left( \frac{l_1}{D}, \frac{l_2}{D}, \frac{l_3}{D}, \frac{l_4}{D}, SR, \frac{P_o}{P_i}, \frac{\rho ND^2}{\mu}, \frac{\epsilon}{D}, \text{blade geometry} \right) \quad (3)$$

$$T_{\text{pump}} = T_{\text{pump}} \left( \frac{l_1}{D}, \frac{l_2}{D}, \frac{l_3}{D}, \frac{l_4}{D}, SR, \frac{P_o}{P_i}, \frac{\rho ND^2}{\mu}, \frac{\epsilon}{D}, \text{blade geometry} \right) \quad (4)$$

where

- SR = speed ratio.
- $P_o/P_i$  = pressure ratio = mean pressure at the outlet/mean pressure at the inlet
- $\rho ND^2/\mu$  = Reynolds number based on the diameter
- $\mu$  = fluid viscosity
- $\epsilon/D$  = surface roughness ratio

Pressure ratio, Reynolds number, and roughness ratio are included to account for the centrifugal and viscous effects. The above expressions lead to the following fundamental relationship of power and torque to speed and diameter:

$$\text{Power}_{\text{pump}} = C_{1,\text{pump}} N_{\text{pump}}^3 D_{\text{pump}}^5 \quad (5)$$

$$\text{Torque}_{\text{pump}} = C_{2,\text{pump}} N_{\text{pump}}^2 D_{\text{pump}}^5 \quad (6)$$

where

$$C_{1,\text{pump}} = \rho^* P_{\text{pump}} = C_{1,\text{pump}} \left( \frac{l_1}{D}, \frac{l_2}{D}, \frac{l_3}{D}, \frac{l_4}{D}, SR, \frac{P_o}{P_i}, \frac{ND^2}{\mu}, \frac{\epsilon}{D}, \text{blade geometry} \right) = \text{capacity factor for input power} \quad (7)$$

$$C_{2,\text{pump}} = \rho^* T_{\text{pump}} = C_{2,\text{pump}} \left( \frac{l_1}{D}, \frac{l_2}{D}, \frac{l_3}{D}, \frac{l_4}{D}, SR, \frac{P_o}{P_i}, \frac{ND^2}{\mu}, \frac{\epsilon}{D}, \text{blade geometry} \right) = \text{capacity factor for input torque} \quad (8)$$

In a family of geometrically similar torque converters with the same roughness ratio,  $l_1/D$ ,  $l_2/D$ ,  $l_3/D$ ,  $l_4/D$ , and  $\epsilon/D$  are constant. For the purpose of this analysis, the effects of Reynolds number are assumed to be relatively small and may be ignored. The functional relationships for geometrically similar torque converters with the same roughness ratio are as follows:

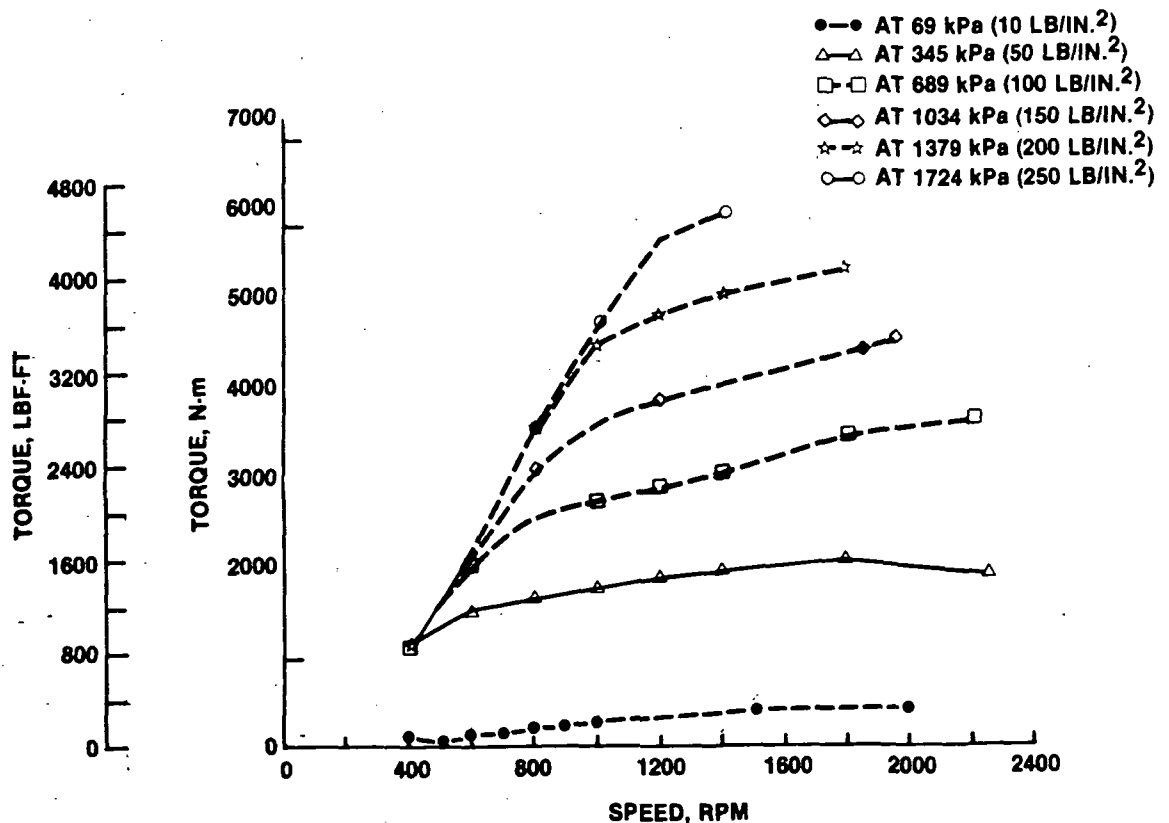
$$C_{1,pump} = \rho * P_{pump} = C_{1,pump} \left( SR, \frac{P_o}{P_i} \right) \quad (9)$$

$$C_{2,pump} = \rho * T_{pump} = C_{2,pump} \left( SR, \frac{P_o}{P_i} \right) \quad (10)$$

The equations (equations 1 through 10) for power and torque are applicable to both the input and output elements. That is, the same type of equations can be derived for the converter turbine element. The formulations are valid as long as the torque converter is fully filled and there is no severe cavitation. The coefficients are generally obtained from experimental data. Since the input power transferred by the torque converter is greatly determined by the pump element, equations 1-10 are generally used for the pump element only. The turbine element is modified to obtain desired torque ratio and efficiency characteristics.

#### Comparison with the Experimental Data

The use of nondimensional analysis has the important practical advantage of collapsing into virtually a single curve results that would otherwise require a multiplicity of curves if plotted dimensionally. Experimental results ob-



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Figure 15. Performance characteristics of a DDA hydraulic retarder--absorption torque as a function of speed and charging pressure.

tained by Detroit Diesel Allison (DDA) are shown in Figure 15 for the case of a hydraulic retarder. Hydraulic retarder is a fluid coupling where the output element is stationary. It acts as an auxiliary braking system to be used on trucks, buses, and tanks. Figure 15 shows the retarder's absorption torque as a function of the impeller's rotational speed and the inlet charging pressure. Several curves are required to represent the effects of rotational speed and inlet charging pressure. The results are also plotted in terms of torque coefficient,  $C_2$ , versus pressure ratio, as shown in Figure 16. It is interesting to see that all the curves collapse into a single curve except for the case of 68.9 kPa (10 lb/in.<sup>2</sup>) of inlet charging pressure. The effect is due to cavitation. Under cavitating flow conditions, dynamical similarity does not exist. The torque coefficient of Figure 16 was obtained for a specific retarder. This experimental data can be applied to a range of different retarder sizes as long as these retarders are geometrically similar and cavitation is not present.

#### Design and Performance Characteristics

The torque converter design and the performance characteristics of the convertible engine torque converter are made based on the actual 0.292 m (11.5 in.) experimental model underdrive-overdrive torque converter of General Motors Advanced Engineering Staff. Dimensions of the toroidal cross section for the baseline torque converter are shown in Figure 17. Its steady-state performance characteristics are shown in Figure 18 for the case of fully filled condition with 552 kPa (80 lb/in.<sup>2</sup>) of inlet charging pressure and 102 N-m (75 lbf-ft)

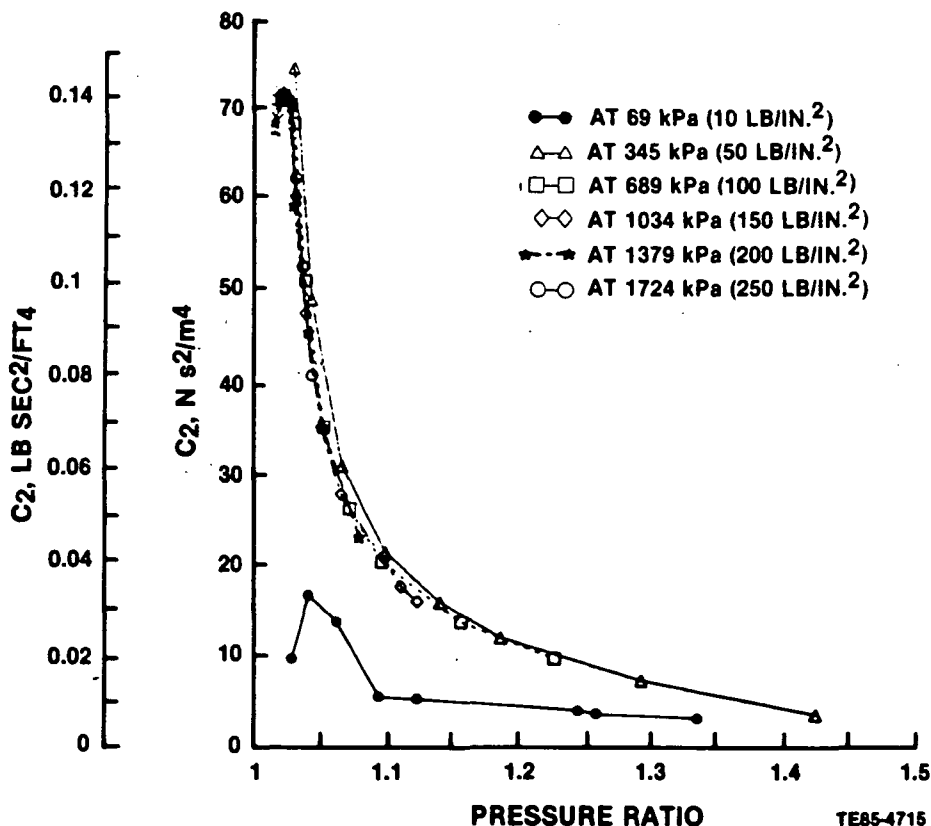
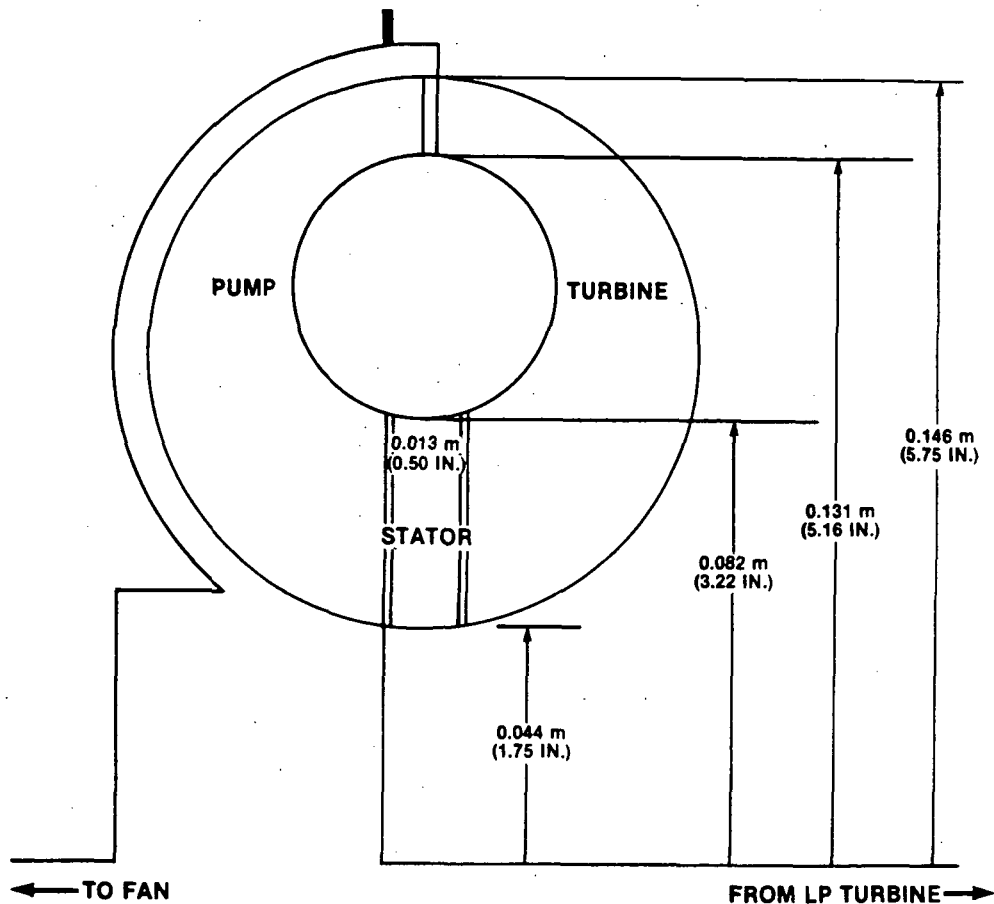


Figure 16. Performance characteristics of a DDA hydraulic retarder-- torque efficient as a function of pressure ratio.



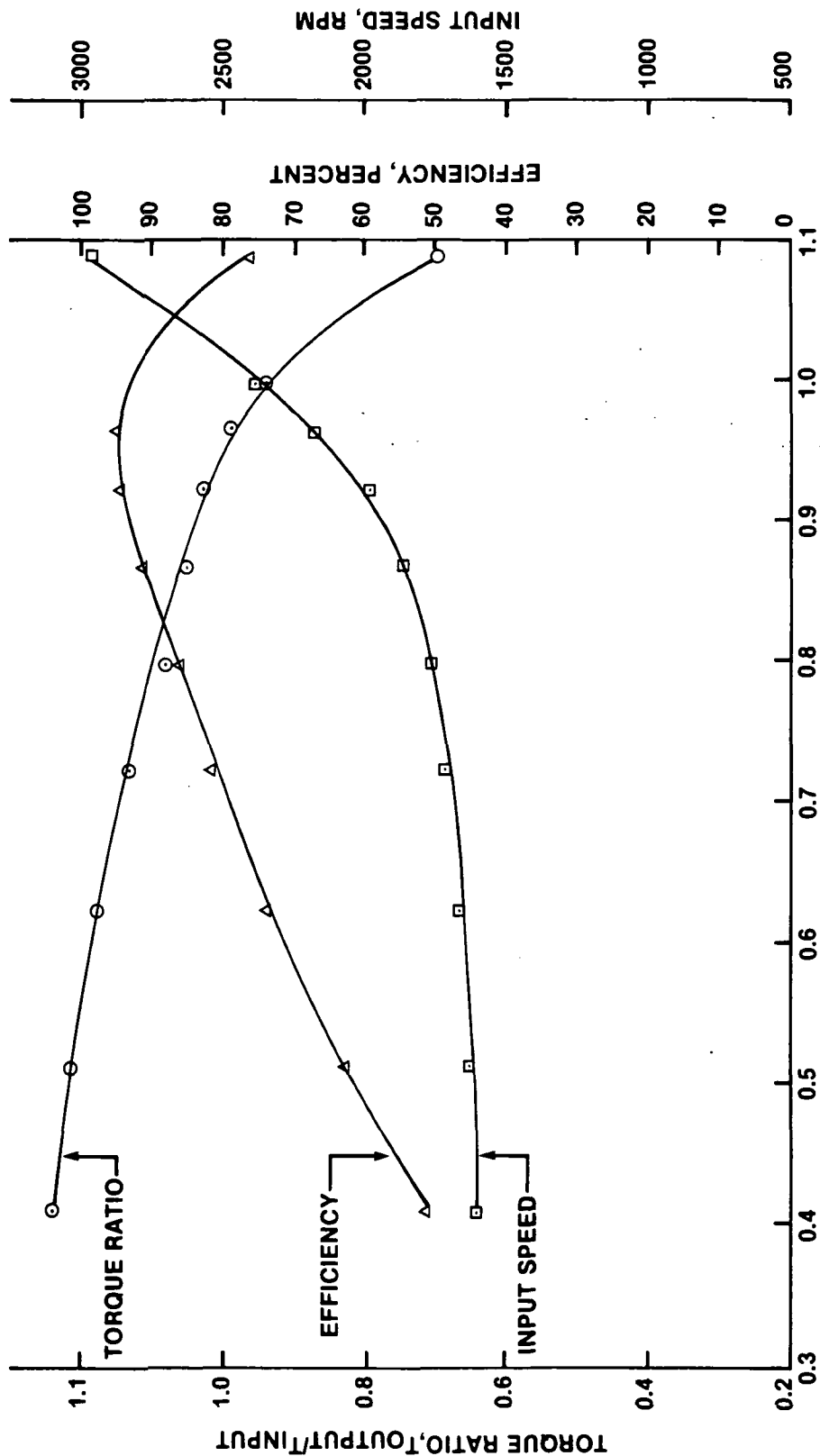
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Figure 17. Toroidal cross section of the baseline 0.292 m (11.5 in.) automotive torque converter.

of input torque. For the convertible engine application, the torque converter needs to deliver maximum power when the speed ratio is close to one. The convertible engine torque converter must be designed to transfer power for this condition.

At the speed ratio of one, the baseline torque converter has the following performance characteristics:

OD	= 0.292 m (11.5 in.)
Charging pressure	= 552 kPa (80 lb/in. <sup>2</sup> )
Input torque	= 102 N-m (75 lbf-ft)
Input speed	= 2400 rpm
Speed ratio	= 1.0
Torque ratio	= 0.94
Efficiency	= 94%



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### SPEED RATIO, $N_{OUT}/N_{INPUT}$

Figure 18. Steady-state performance characteristics of the 0.292 m (11.5 in.) baseline automotive torque converter with 102 N-m (75 lbf-ft) of constant torque.

At the speed ratio of one, the maximum design requirements for the convertible engine torque converter, in the converter mode, are the following:

Input torque	= 5077 N-m (3744 lbf-ft)
Input speed	= 6986 rpm
Speed ratio	= 1.0

In reference to the formulation for scaling of equations (6) and (10), the torque converter's diameter required to transfer the torque for these design conditions is 0.416 m (16.4 in.) if the torque converter inlet charging pressure is 9549 kPa (1385 lb/in.<sup>2</sup>).

Dimensions of the toroidal cross section corresponding to the 9549 kPa (1385 lb/in.<sup>2</sup>) of inlet charging pressure is shown in Figure 19. However, it is totally unrealistic to supply 9549 kPa (1385 lb/in.<sup>2</sup>) of charging pressure to the torque converter system. It is more practical to design the oil system with the capability of providing about 1379 kPa (200 lb/in.<sup>2</sup>) of charging pressure. For this reason, the charging pressure is chosen to be 1379 kPa (200 lb/in.<sup>2</sup>) at the design operating point. It can be seen from Figure 15

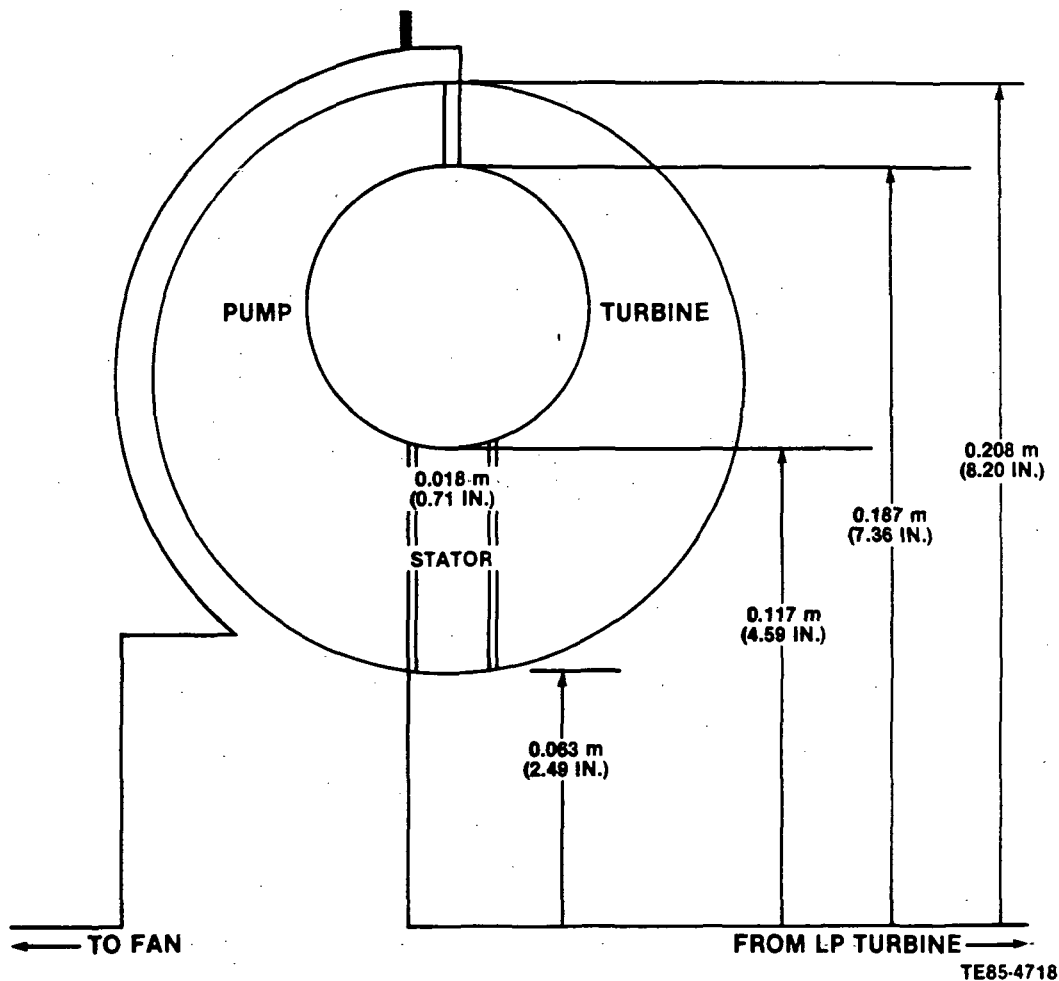
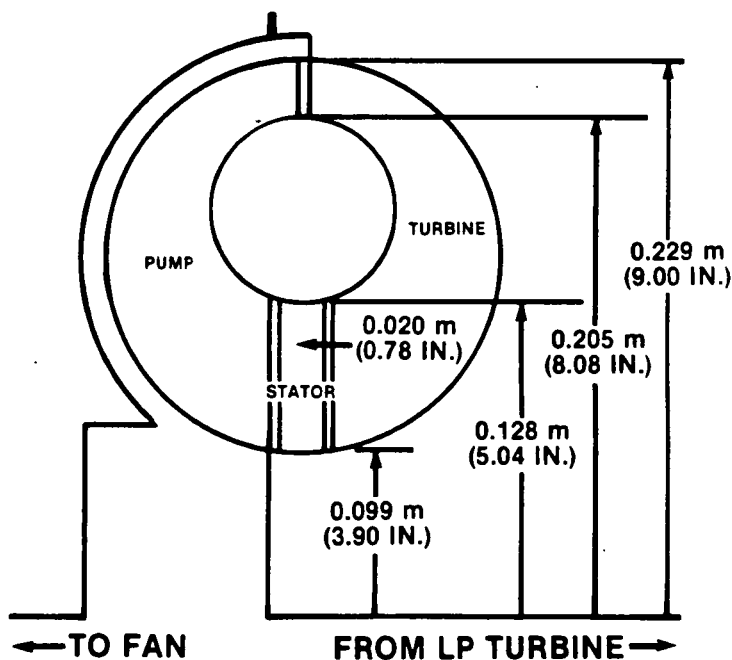


Figure 19. Toroidal cross section of the convertible engine torque converter with  $P_{\text{charging}} = 9549 \text{ kPa}$  (1385 lb/in.<sup>2</sup>).



that the inlet charging pressure has a tremendous effect on the ability of the torque converter to transfer the torque. Since the inlet charging pressure is reduced from 9549 kPa (1385 lb/in.<sup>2</sup>) to about 1379 kPa (200 lb/in.<sup>2</sup>), the 0.417 m (16.4 in.) torque converter cannot transfer 5077 N-m (3744 lbf-ft) of input torque at 6986 rpm of input speed and a speed ratio of one. Accurate prediction cannot be made to estimate the effect of charging pressure on torque in the range of 1379 to 9549 kPa (200 to 1385 lb/in.<sup>2</sup>) since all the available experimental data are at much lower charging pressure, speed, torque, and power. Based on the limited amount of experimental data, the diameter required to transfer 5077 N-m (3744 lbf-ft) (at 6986 rpm, speed ratio of one, and 1379 kPa [200 lb/in.<sup>2</sup>] of charging pressure) is predicted to be 0.451 m (18.0 in.). Dimensions of the toroidal cross section corresponding to the 1379 kPa (200 lb/in.<sup>2</sup>) of inlet charging pressure is shown in Figure 20. Steady-state performance characteristics of the convertible engine torque converter are shown in Figure 21 for the case of a fully filled condition with 1379 kPa (200 lb/in.<sup>2</sup>) of charging pressure and 6986 rpm of input speed.

It was mentioned that the convertible engine torque converter operated under several percent fill conditions. The torque converter is empty at the beginning of the transition from shaft mode to turbofan mode. In addition, the torque converter inlet charging pressure is basically zero during the partially filled conditions. Since the torque converter's performance characteristics of Figure 21 are applicable only to a fully filled torque converter with 1379 kPa (200 lb/in.<sup>2</sup>) of charging pressure, two additional correction factors are required to simulate the performance characteristics of the convertible engine. The two correction factors are used to correct for the effect of percent fill and the effect of charging pressure.



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Figure 20. Toroidal cross section of the convertible engine torque converter with  $P_{\text{charging}} = 1379 \text{ kPa}$  (200 lb/in.<sup>2</sup>).

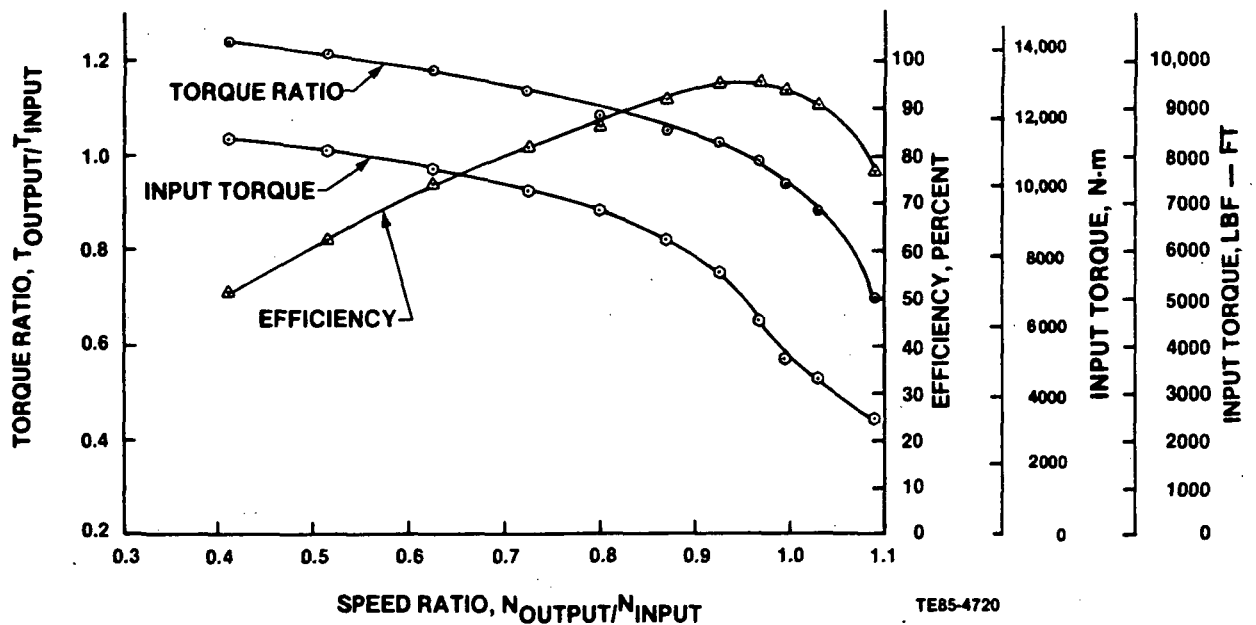


Figure 21. The assumed steady-state performance characteristics of the fully filled overdrive-underdrive torque converter at 6987 rpm--1379 kPa (200 lb/in.<sup>2</sup>) inlet pressure.

#### Correction Factor for the Effect of Percent Fill on Torque

Performance characteristics of the partially filled torque converter are one of the most difficult to analyze since the flow is highly unsteady and three dimensional. A very limited amount of work has been done in this area at much lower speed and power levels. Two important phenomena are associated with the partially filled torque converter:

1. At low partial filled levels, the oil circulates in the toroidal circuit in the same manner as in the case of a fluid coupling. That is, the flow circulates from pump to turbine and from turbine back to pump. In the coupling phase, the torque ratio is one. The control of the oil level in the filling and dumping modes will be semicritical as we have determined that as a converter is being filled and evacuated, its capacity for transferring torque will begin to diminish in an orderly progression as would normally be expected, but at some partially filled or dumped level, capacity will again rise very rapidly. This is because at some level of operation, the reduced amount of oil that is left in the toroidal circuit begins to establish a flow path in the area of the pump exit and the turbine inlet and almost instantaneously provides the torque capacity of a hydrodynamic fluid coupling operating at a much larger diameter fluid flow path. At high partially filled levels, the oil circulates in the same manner as in the case of a torque converter where the torque ratio is greater than one. Figure 22 illustrates this phenomenon.

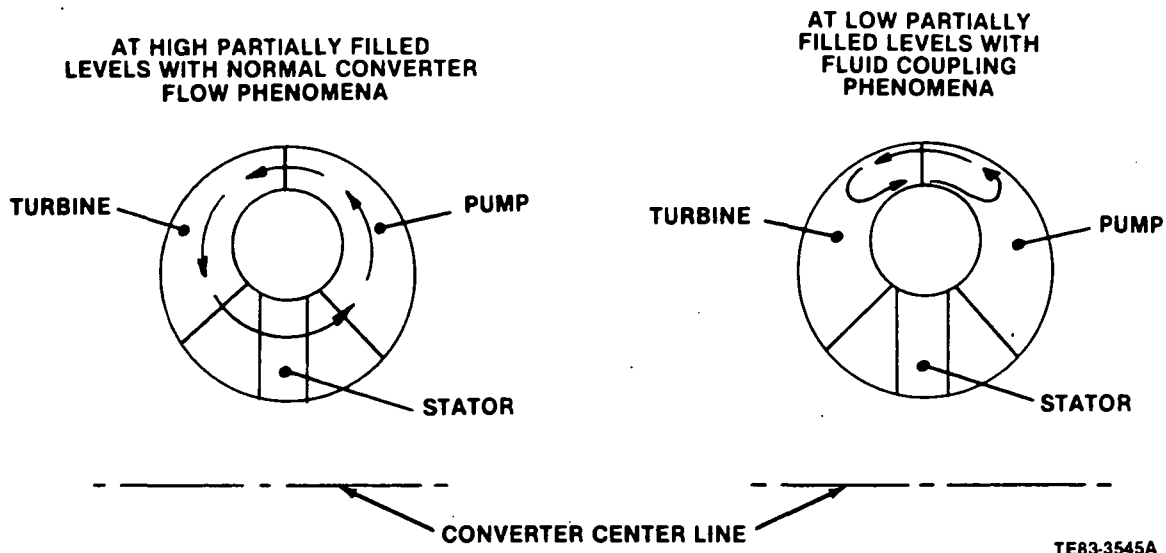


Figure 22. Torque converter internal flow path.

2. Work on a partially filled fluid coupling and torque converter was studied by Japanese investigators. Technical details of the work were presented in JSME paper No. 781, dated 8 August 1968, entitled "Characteristics of Fluid Couplings." In reference to the paper, the coupling characteristics occur in the range of high speed ratios and the converter characteristics occur in the range of low speed ratios. In the case of a fully filled condition, torque characteristics are not affected significantly by the oil temperature, while in the case of partially filled conditions, torque characteristics vary widely with the change of the oil temperature.

The air-liquid mixture is affected by the physical and/or chemical properties of the working liquid. At high temperatures, the air-liquid mixture occupies the whole toroidal cavity. It was concluded that there were four types of flow patterns:

- o The first type of flow occurs when the speed ratio is between 0.8 and 1.0. Air is separated from the working fluid that is concentrated to the outer portion of the toroidal cavity.
- o The second type of flow occurs when the speed ratio is between 0.6 and 0.8. A portion of the fluid concentrates on the inner radius of the toroidal cavity and the fluid contains small air bubbles. The mass of air remains at the inner radius of the toroidal cavity.
- o The third type of flow occurs when the speed ratio is between 0.4 and 0.6. The air-liquid mixture circulates in the toroidal cavity as a whole. The mass of air remains in the central portion of the cavity.
- o The fourth type of flow occurs when the speed ratio is between 0 and 0.4. In this range of speed ratio, air-liquid mixture circulates in the toroidal cavity as a whole without the mass of air.

The effect of partially filled volume on torque was estimated based on the available experimental data from Detroit Diesel Allison and from the above JSME paper. The effect of percent fill on torque is shown in Figure 23. Due to the limited amount of experimental data, it is assumed in this analysis that the effect is independent of speed ratio. Results are presented in terms of correction factor for torque, as a function of speed ratio, and percent fill. The correction factor for torque is normalized with respect to the torque at fully filled condition.

#### Correction Factor for the Effect of Inlet Charging Pressure

The experimental data of the hydraulic retarder (Figure 15) shows that the inlet charging pressure has a significant effect on torque. Charging pressure is often required in torque converter and fluid coupling to avoid cavitation that occurs when the static pressure is less than the vapor pressure of the fluid. Cavitation is most severe at stall (speed ratio = zero) where the efficiency is zero and all the inlet power is dissipated into heat. Charging pressures used in the current automotive applications vary from 207 to 1724 kPa (30 to 250 lb/in.<sup>2</sup>). In addition to the phenomenon associated with cavitation, charging pressure also has an effect on the velocity and pressure distributions around the blade within the blade passages. The effect of charging pressure on torque is shown in Figure 24. Results are presented in terms of correction factor for torque as a function of speed ratio and charging pressure. Since 1379 kPa (200 lb/in.<sup>2</sup>) was chosen as the charging pressure at the operating design point, the correction factor for torque was normalized

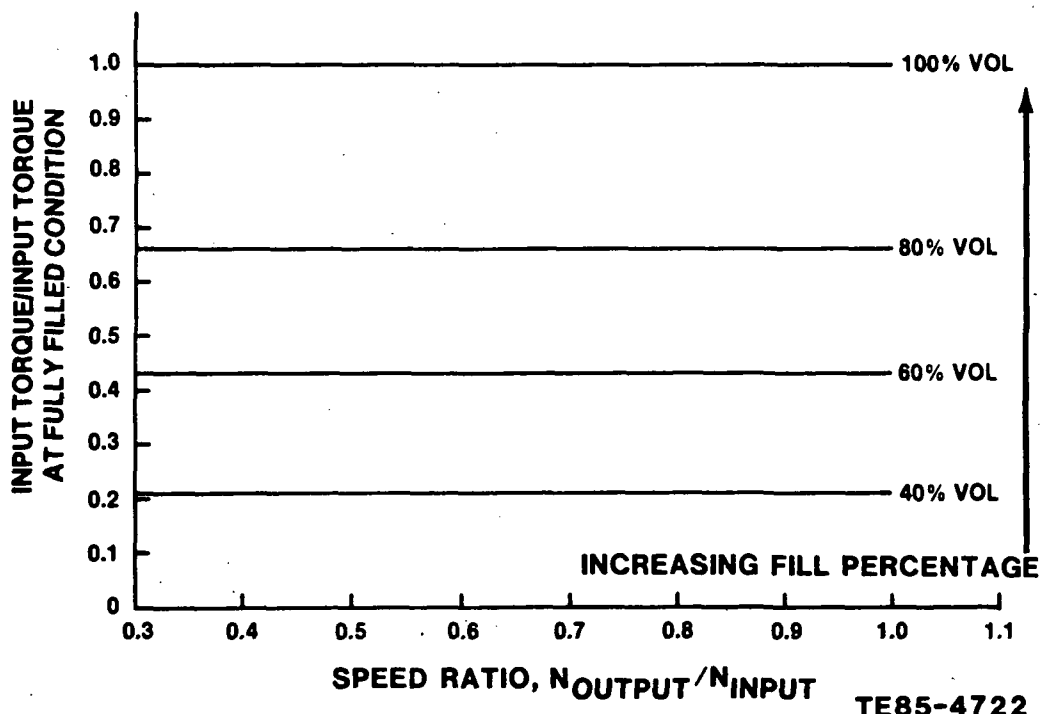


Figure 23. Qualitative trend to indicate the effect of partially filled volume on input torque (based on limited amount of experimental data).

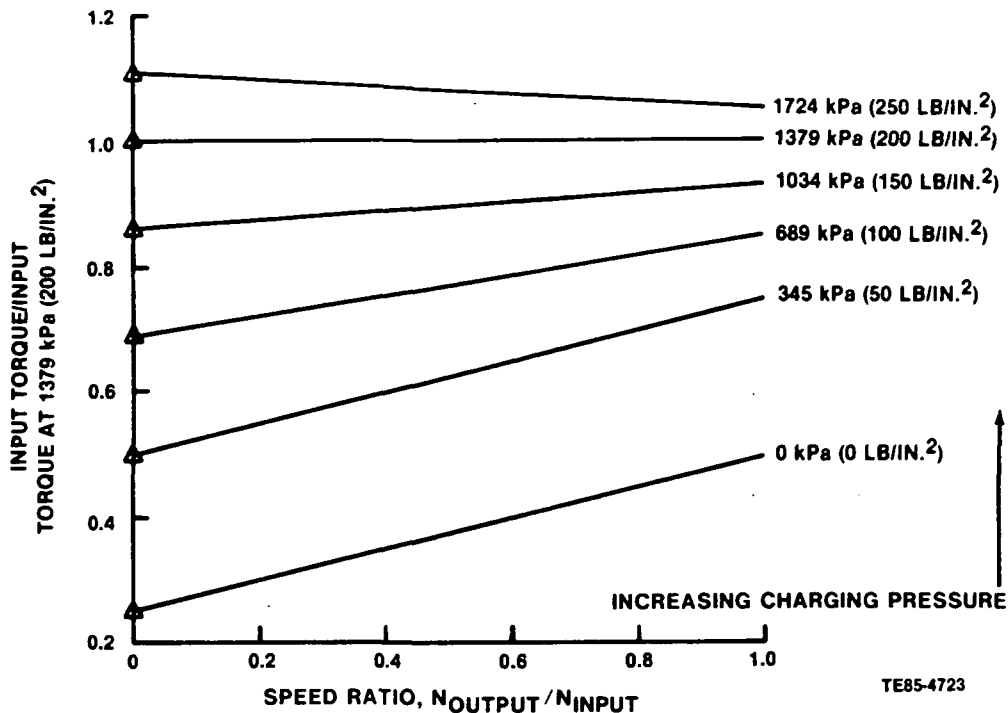


Figure 24. Qualitative trend to indicate the effect of charging pressure on input torque based on experimental data of Hydramatic and Detroit Diesel Allison Division, General Motors Corporation.

with respect to the torque at 1379 kPa (200 lb/in.<sup>2</sup>) of charging pressure. Figure 24 shows that the effect of charging pressure is more effective at lower speed ratios. This is because cavitation is more severe at lower speed ratios and thus more potential for improvement exists. The effect of charging pressure was estimated based on experimental data of Hydramatic and Detroit Diesel Allison.

The torque converter inlet charging pressure can be used to control the acceleration of the fan and to synchronize the fan and power turbine speeds for the following reasons:

- o The charging pressure has a very significant effect on the torque converter input torque as shown in Figure 24. Thus, an increase in charging pressure increases torque converter input power at a given input speed.
- o The power transmitted to the input section of the torque converter equals the power transferred to the fan and the torque converter power losses. Since the charging pressure does not have an adverse effect on the torque converter efficiency, the charging pressure also has a favorable effect on the fan output power. The fan speed increases as a result of the increase in the fan output power.

The following important facts must be mentioned:

1. The correction factors for partial fill and for charging pressure of Figures 23 and 24 were estimated based on the limited amount of available data at much lower speed and power than that encountered in the convertible engine torque converter's application. Therefore, these correction factors are intended to provide a qualitative trend to indicate the effects of partially filled volume and torque converter inlet charging pressure. Experimental models must be built and tested to determine the actual performance characteristics and the two correction factors for the convertible engine torque converter.
2. The steady-state performance characteristics of Figure 21 along with the correction factors of Figures 23 and 24 are used to analyze the non-steady-state phenomenon for the convertible engine torque converter. The real transient effect, in terms of delay time for response, is not included in the analysis since no experimental data are available. Therefore, tests must be performed to determine the actual transient effect.

#### PRINCIPLE AND OPERATION

A simplified schematic diagram of the convertible engine torque converter is shown in Figure 25. The power generated by the high pressure (HP) turbine is used to drive the compressor. The power generated by the low pressure (LP) turbine is given to the prop rotor (through a clutch device) and the input pump element of the torque converter. The power from the LP turbine is given solely to the prop rotor during takeoff and landing. In the convertible engine fan drive application, the torque converter is empty at the beginning of the transition from shaft mode to turbofan mode, and the fan is windmilling. When

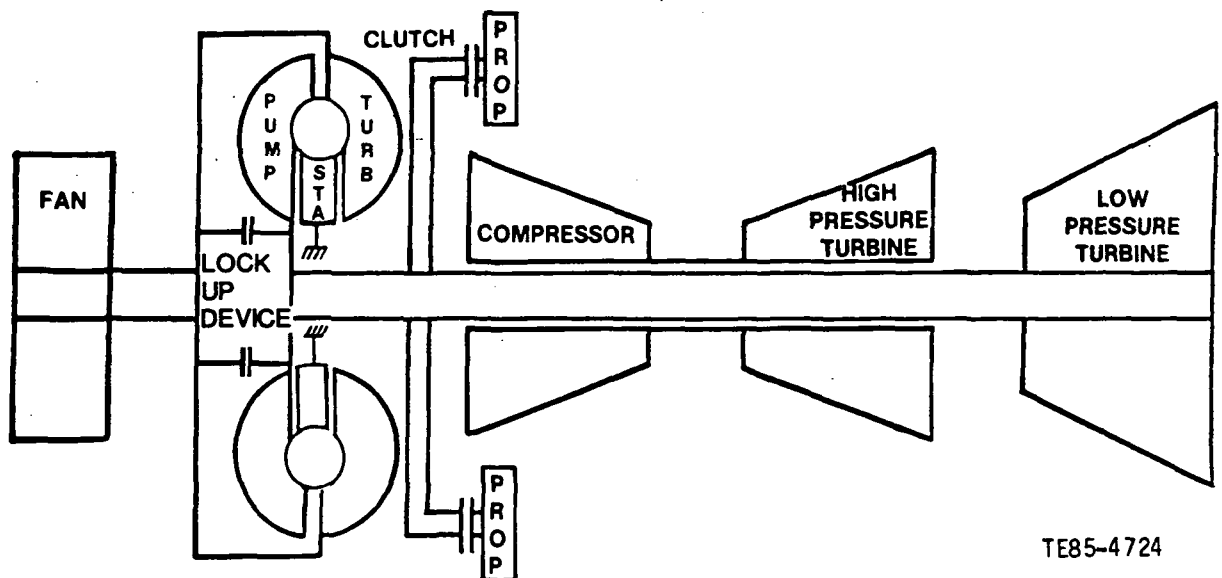


Figure 25. Simplified schematic diagram of convertible engine torque converter.

power transfer to the fan is initiated, the oil is pumped into the torque converter toroidal cavity. As the volume of oil inside the torque converter increases, more power is transferred to the torque converter and the fan accelerates. The fan continues to accelerate until a one-to-one speed ratio is obtained between the fan speed and the LP turbine speed. The mechanical lockup device is activated when a speed ratio of one to one is reached. To minimize losses, the oil is dumped out of the torque converter after the engagement of the mechanical lockup device.

#### Power Transfer Cycle from Proprotor Mode to Fan Mode--Performance Analysis

A performance simulation model was developed to analyze the overall performance of the convertible engine torque converter during the power transfer cycle from the prop mode to the fan mode. Suitable filling and dumping curves along with the performance characteristics of the torque converter were included in the model. The model was based on the following physical facts: (1) the power generated by the LP turbine is transmitted to the proprotor (until it is decoupled) and to the input section of the torque converter, (2) the power transmitted to the input section of the torque converter equals the power transferred to the fan and the power losses inside the torque converter, and (3) the power transferred to the fan consists of the power required by the fan for constant rotation and the excess power for acceleration. This excess power for fan acceleration determines the time it takes the fan to reach the LP turbine speed.

The computer model simulating the convertible engine conversion from shaft to fan power was used to calculate engine data during transients at the design condition of SL/370 km/h (200 KTAS) with a torque converter fill time of 6 sec. Conversions are initiated for the fold tilt rotor aircraft at a proprotor speed of 80%. Engine power turbine speed and proprotor speed (until disengagement) are held constant through the conversion. Proprotor thrust is reduced as fan thrust increases to maintain constant total thrust until the prop is no longer supplying propulsion. At this point, thrust can be allowed to increase as the fan continues to accelerate to 80% rpm, or the proprotor can be decoupled and thrust held constant by permitting the power turbine to reduce speed and match fan speed. In either case, fan and power turbine shaft speeds will be matched and mechanically engaged.

The torque converter inlet charging pressure and the gas turbine temperature are the main parameters used to control the acceleration of the fan. The charging pressure has a significant effect on the torque converter input and output torques. In the first performance model, the inlet charging pressure curve of Figure 26 was used. The inlet charging pressure was basically zero during the first 6 sec of filling time. It was then increased linearly to 1379 kPa (200 lb/in.<sup>2</sup>) in 0.5 sec. It was found that this type of increase in charging pressure may provide the ability for the torque converter to transfer more power than the engine can generate. As a result, the fan may accelerate too fast and the gas turbine temperature may reach the limiting value. Another performance model was made based on the charging pressure curve of Figure 27. The charging pressure was zero during the first 6 sec; it was then increased linearly to 276 kPa (40 lb/in.<sup>2</sup>) for the next 0.5 sec. After that it was increased linearly to 1379 kPa (200 lb/in.<sup>2</sup>) in 0.2 sec. It was then con-

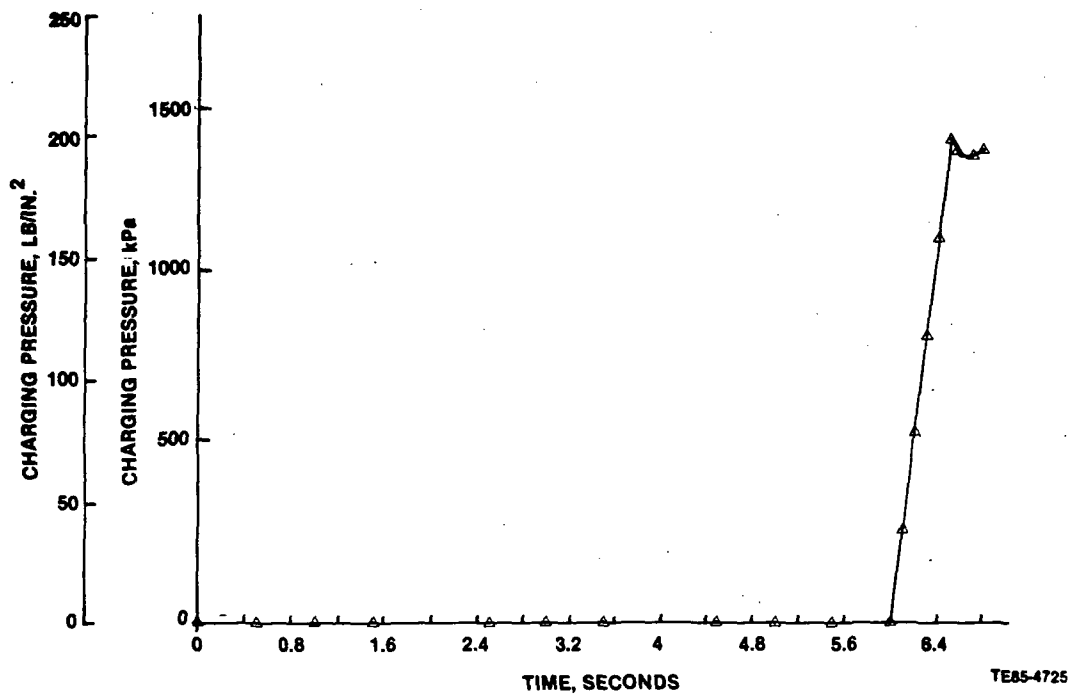


Figure 26. Performance characteristics of the convertible engine torque converter (filling mode)---original torque converter charging pressure schedule.

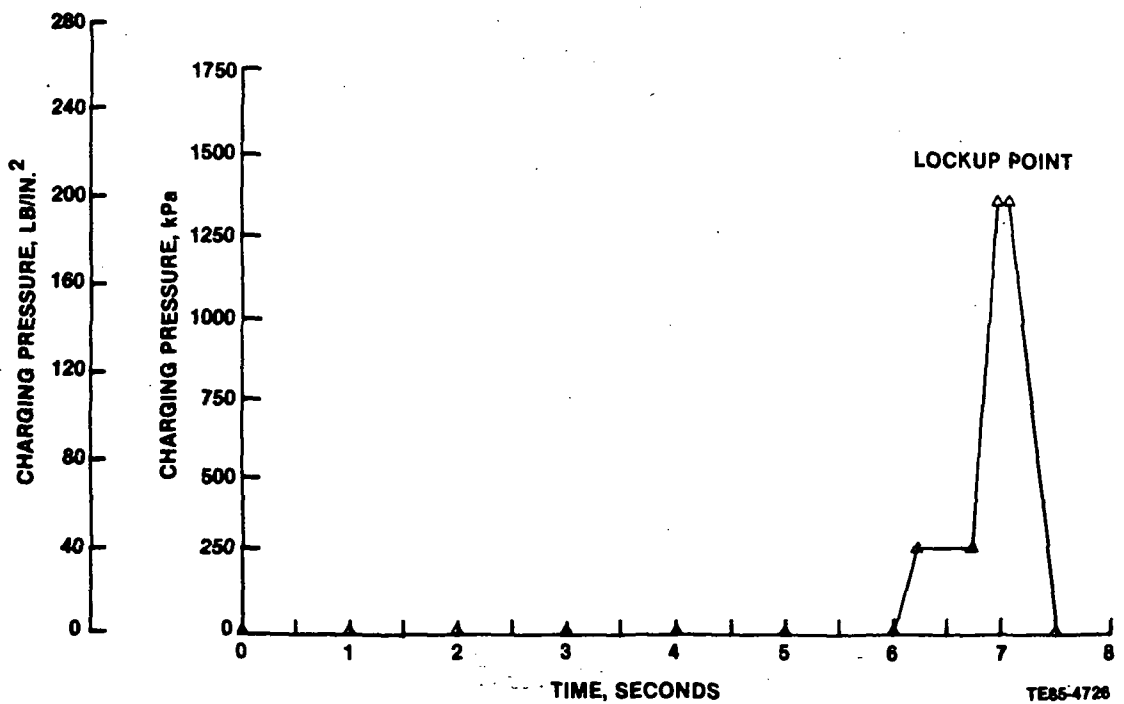


Figure 27. Performance characteristics of the convertible engine torque converter (filling mode)---improved torque converter charging pressure schedule.



trolled around 1379 kPa (200 lb/in.<sup>2</sup>) until a one-to-one speed ratio between the fan speed and the LP turbine speed was obtained. After the mechanical coupling was engaged, the charging pressure was decreased linearly from 1379 kPa (200 lb/in.<sup>2</sup>) to 0 in 0.5 sec. The dumping process was then initiated at a rate of 1.262 L/s (20 gal/min) through the exhaust holes.

The filling and percent torque curves are shown as a function of time in Figure 28. The percent torque is a parameter used to indicate the effects of partially filled volume and torque converter inlet charging pressure. The torque converter performance characteristics in terms of speed ratio and torque ratio are shown as a function of time in Figure 29. The power generated by the low pressure turbine, along with the breakdown of the power given to the propotor and to the input section of the torque converter, are given as a function of time in Figure 30. Figure 31 shows the power losses in the torque converter and the fan excess power for acceleration. Figure 32 shows the fan speed relative to the LP turbine speed. It is seen that the fan speed reaches the LP turbine speed at 1.0 sec after the torque converter is completely filled. The gas temperature is shown as a function of time in Figure 33.

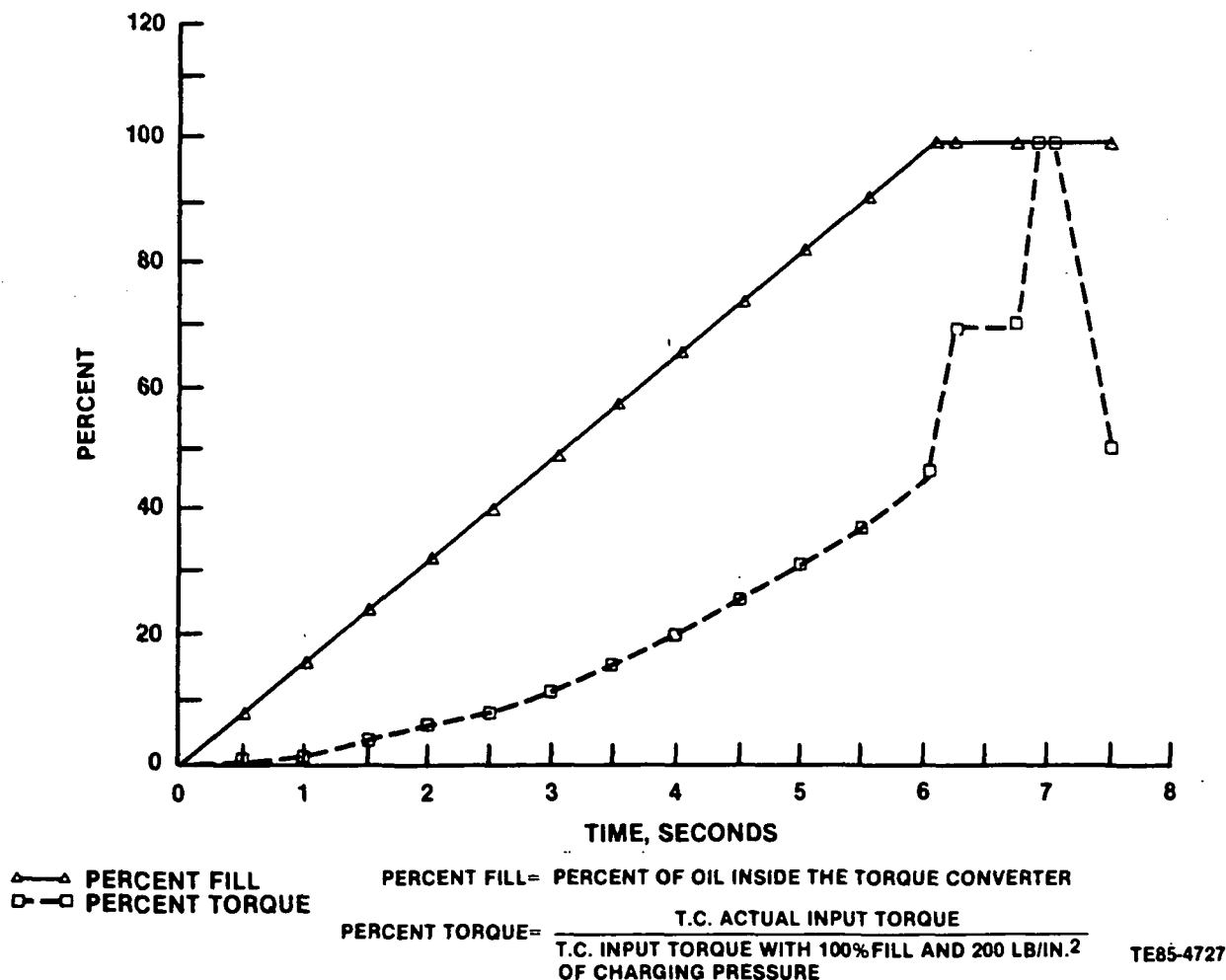


Figure 28. Performance characteristics of the convertible engine torque converter (filling mode)--percent torque and percent fill curves.

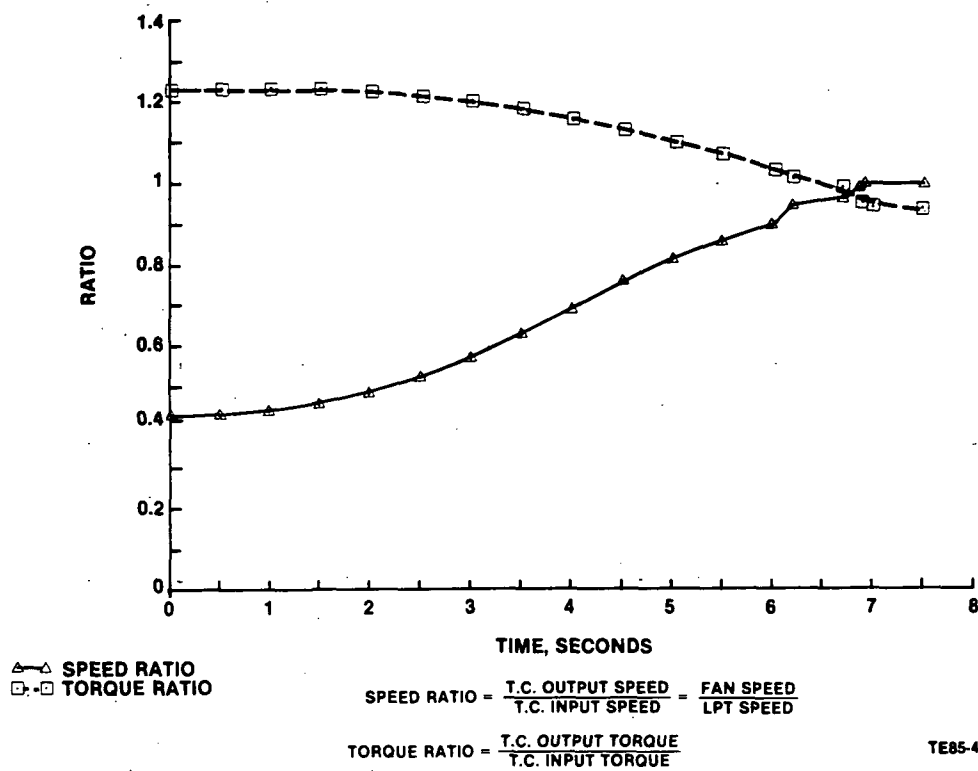


Figure 29. Performance characteristics of the convertible engine torque converter (filling mode)--speed ratio and torque ratio curves.

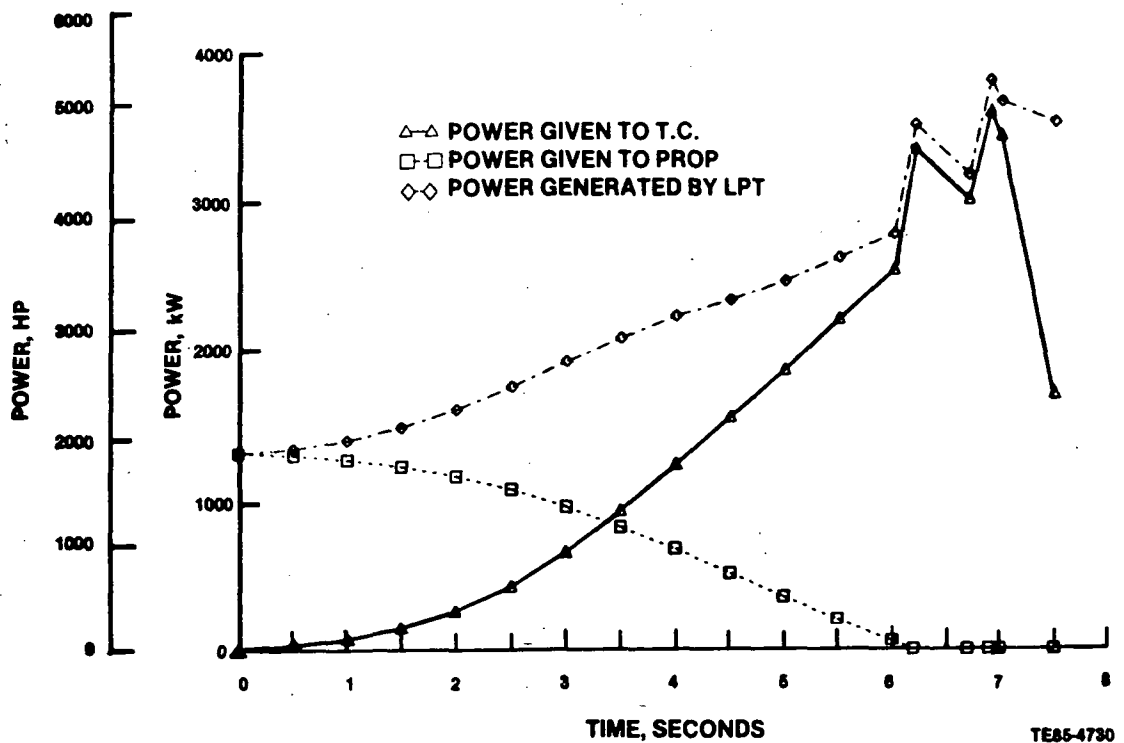


Figure 30. Performance characteristics of the convertible engine torque converter (filling mode)--distribution of power from the LP turbine to the prop and the torque converter.

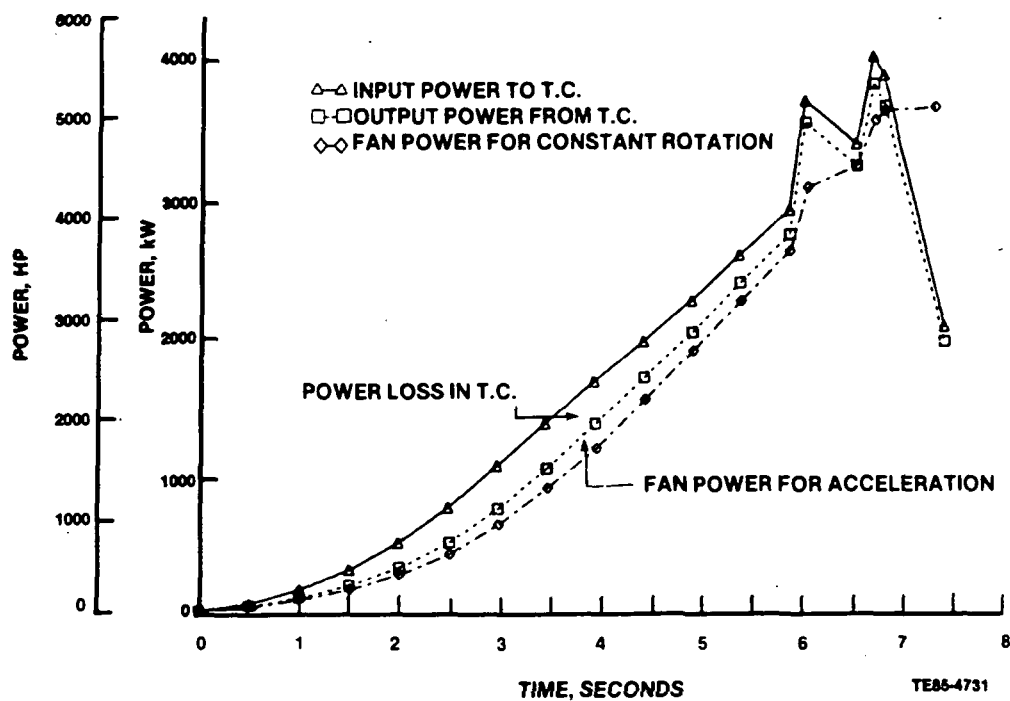


Figure 31. Performance characteristics of the convertible engine torque converter (filling mode)--power loss in torque converter and the excess power for fan acceleration.

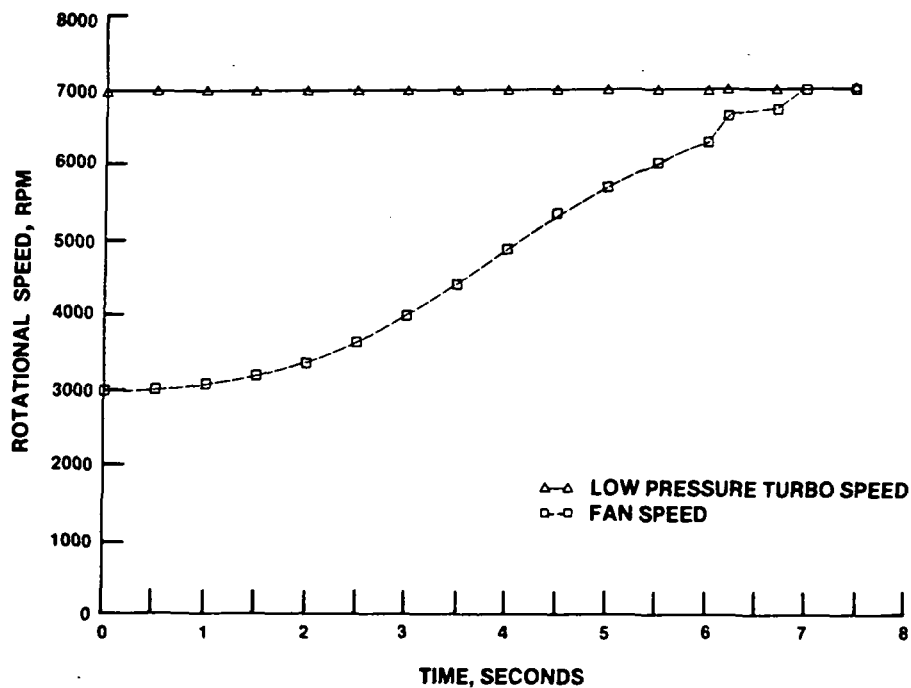


Figure 32. Performance characteristics of the convertible engine torque converter (filling mode)--acceleration of the fan speed.

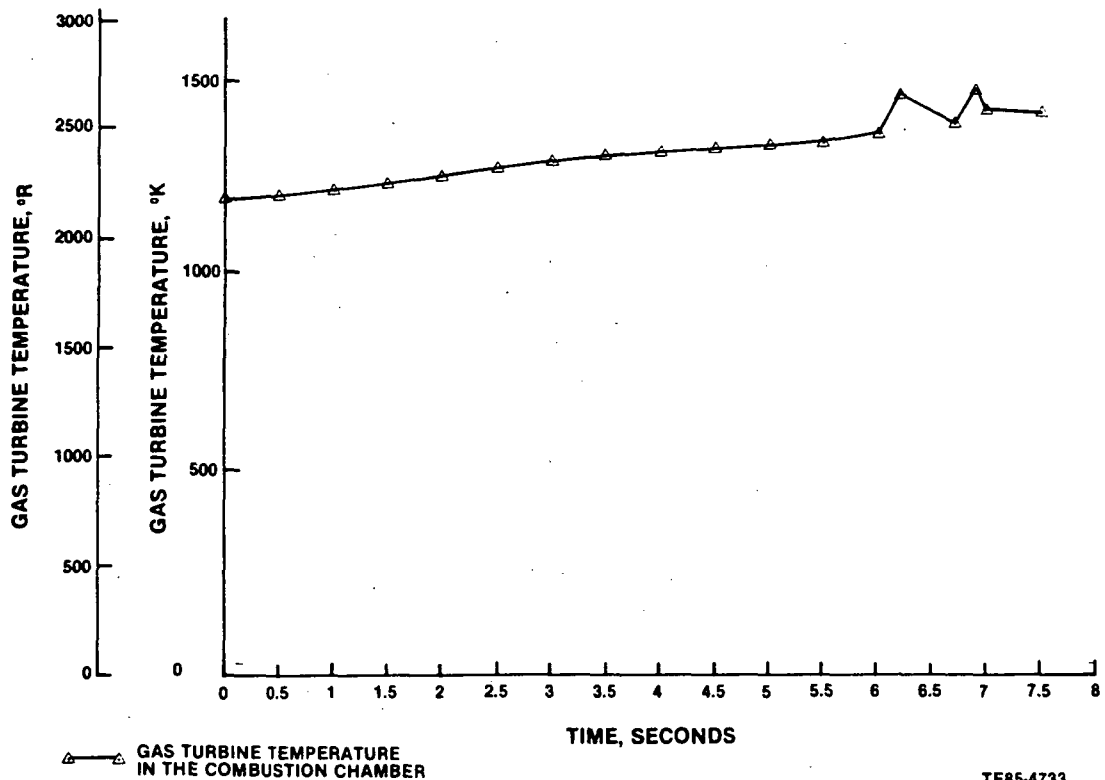


Figure 33. Performance characteristics of the convertible engine torque converter (filling mode)--gas turbine temperature.

#### Power Transfer Cycle from Proprotor Mode to Fan Mode--Heat Rejection Analysis

An investigation was made to determine the torque converter power loss during the power transfer from the prop to the fan mode. The power loss due to the inefficiency of the torque converter causes the oil temperature to increase. The analysis was made to determine if an additional cooling system was required to absorb the power loss generated by the torque converter. Figure 34 shows the torque converter power loss as a function of time. The power loss can be classified into the following three categories:

1. During the first 6 sec of filling time, the energy loss is 1237 kJ (1173 Btu). During this period, there are 22.71 L (6 gal) of oil flowing into the torque converter. The associated increase in oil temperature based on this 22.71 L (6 gal) of oil is 27°K (49°R).
2. During the next 1 sec of lockup time, the energy loss is 172 kJ (163 Btu). In addition to the 15.14 L (4 gal) of oil retained in the torque converter, there is 1.25 L (0.33 gal) of oil leaving the torque converter through the exhaust holes. The associated increase in oil temperature, based on the 16.39 L (4.33 gal) of oil, is 6°K (10°R).
3. During the next 12.5 sec of draining time, the energy loss is 535 kJ (507 Btu). The associated increase in oil temperature, based on the 15.14 L (4 gal) of oil draining out of the torque converter, is 18°K (32°R).

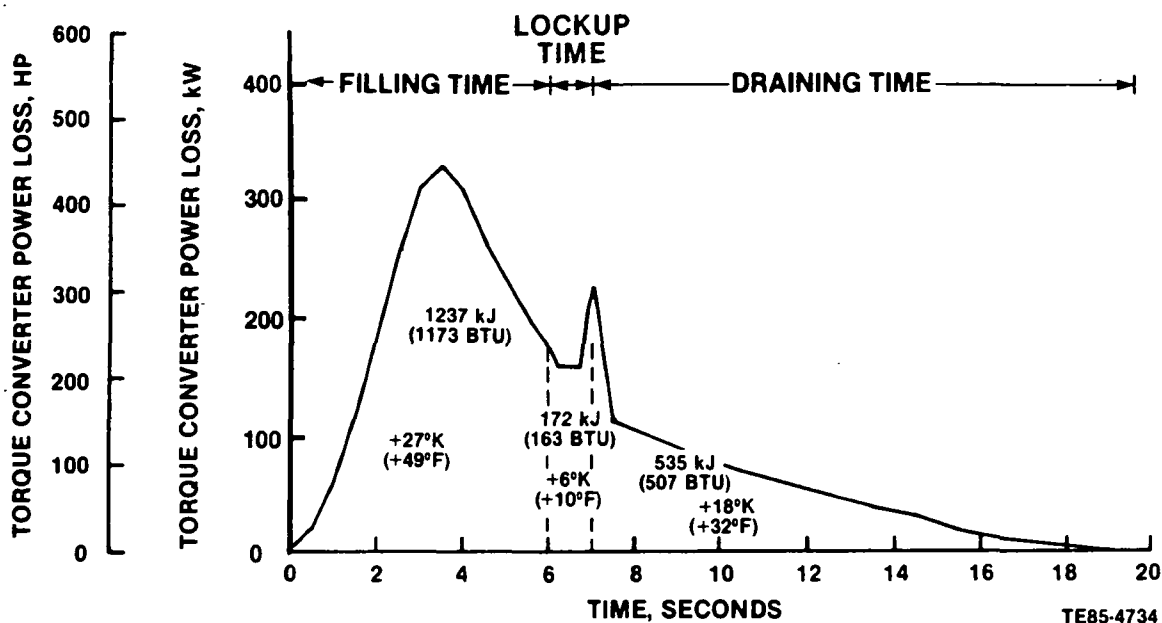
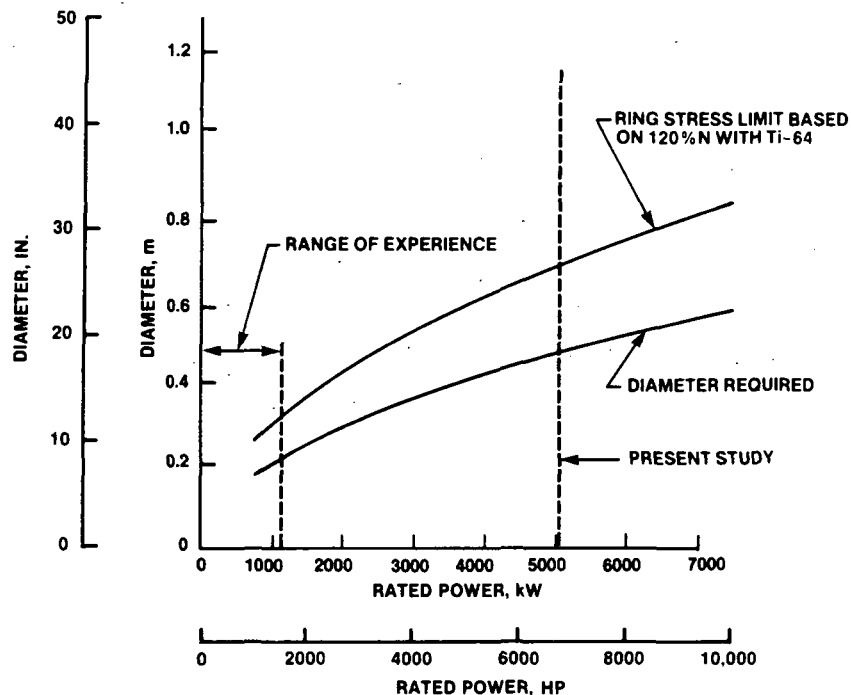


Figure 34. Torque converter power loss.

It is assumed in the calculation of the increase in oil temperature that all of the energy loss is absorbed by the oil. Since part of the energy loss can be transferred by conduction and radiation, the predicted rise in oil temperature tends to be conservative (i.e., higher than expected). The total torque converter energy loss during the power transfer from prop to fan mode is 1944 kJ (1843 Btu). The associated change in oil temperature is about 51°K (91°R). Based on this analysis, it is concluded that a separate cooling system is not required for the torque converter. The engine cooling system can be designed to absorb the additional energy loss associated with the inefficiency of the torque converter.

#### Torque Converter Scaling for the Convertible Engine Application

The required torque converter's diameter is plotted as a function of rated power in Figure 35 based on tip speed at 161 m/s (560 ft/sec) as used in the convertible engine torque converter conceptual design. The allowable diameter based on 120% speed ring stress limit with Ti 6-4 material is also shown in Figure 35. It is seen that the required diameter is smaller than the allowable diameter in the range of 745 to 7450 kW (1000 to 10,000 shp). As the power increases, the difference between the allowable and required diameter becomes larger. This is because the ring stress is proportional to the square power of the diameter and the torque converter power is proportional to the fifth power of the diameter. Therefore, it can preliminarily be concluded that there is no fundamental sizing problem in using the torque converter for the convertible engine application in the range of 745 to 7456 kW (1000 to 10,000 shp) provided that the torque converter's blade tip speed can be increased from 73 to 171 m/s (240 to 560 ft/sec).



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Figure 35. Torque converter scaling for the convertible engine.

The torque converter's blade tip speed required for the convertible engine torque converter application is about 171 m/s (560 ft/sec). The maximum torque converter's blade tip speed used in the automotive application is only 73 m/s (240 ft/sec). Furthermore, aluminum casting and sheet metal stamping are the only processes used to manufacture torque converters for the automotive applications. Titanium forging and investment casting processes are required for the convertible engine torque converter. Fundamental research and development work is needed in order to expand the torque converter's technology from 73 to 171 m/s (240 to 560 ft/sec) of tip speed.

## VIII. PROBLEM IDENTIFICATION

### AXIAL THRUST

Axial thrust encountered within a torque converter can be a severe problem in large units. For example, axial thrust encountered in a 1119 kW (1500 hp) unit (with blade tip speed of 56 m/s [185 ft/sec]) can be as high as 44,482 N (10,000 lb). Since the torque converter is operating under a wide range of speed ratios (from zero to one), axial thrust is determined as a function of speed ratio. Experimental data have shown that maximum axial thrust occurs at stall in small units, but it occurs at high speed ratios in large units.

Thrust encountered within a torque converter is very similar to that in the centrifugal turbomachinery. It is the result of centrifugal pressure loading and the change in axial flow momentum generated by the torque converter element. A free body diagram along with the three axial forces acting on the converter pump is shown in Figure 36. Part of the centrifugal pressure inside the torus is converted to circulating flow motion that results in a net axial force,  $F_1$ , caused by the change in axial flow momentum. The remaining centrifugal pressure inside the torus results in a net axial force,  $F_2$ . The centrifugal pressure also exists in the region between the outside shell of the converter turbine and the converter pump casing. Since there is practically no circulating flow motion, the trapped oil in this region has very high centrifugal pressure. In addition, the area on which this pressure acts is large. Therefore, the axial force due to high centrifugal pressure,  $F_3$ , may be the single most important component of axial thrust since its magnitude is

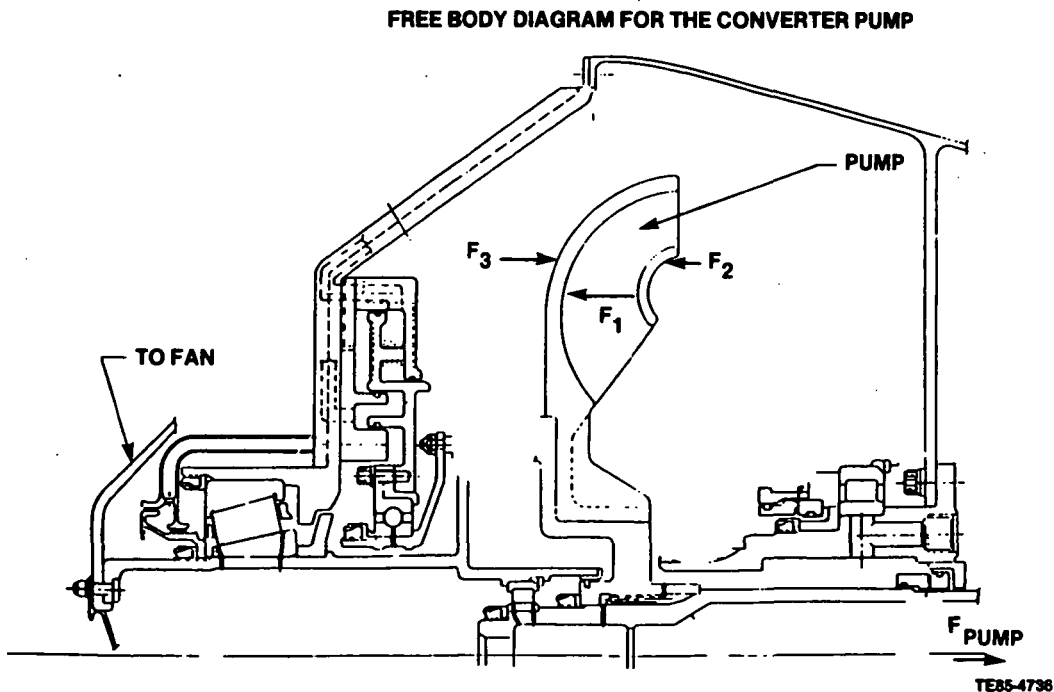


Figure 36. Axial thrust encountered within the torque converter free body diagram for the converter pump.

very high relative to  $F_1$  and  $F_2$ . As a result, a net axial force,  $F_{\text{pump}}$ , pushes the pump element to the right. Similarly, a net axial force,  $F_{\text{turbine}}$ , pushes the turbine element to the left. The two axial forces,  $F_{\text{pump}}$  and  $F_{\text{turbine}}$ , are shown in Figure 37. In the convertible engine torque converter, the forces  $F_{\text{pump}}$  and  $F_{\text{turbine}}$  can be as high as 667,230 N (150,000 lb).

However, the allowable total thrust that the engine can absorb is only about 44,482 N (10,000 lb). Suitable steps must be taken to reduce the torque converter axial thrust to an acceptable value. There are three effective ways of reducing axial thrust. First, the high centrifugal pressure in the oil-trapped region can be reduced by installing short radial blades, as shown in Figure 37. The presence of the blades causes the oil to circulate in the same manner as in the case of a fluid coupling. However, since the blade's aspect ratio (ratio of the blade height over chord) is extremely poor, most of the high pressure in the oil-trapped region is lost due to the effects of secondary flow, three dimensional boundary layers, and incidence losses. On the other hand, the pressure loss in the oil-trapped region also causes a reduction in the torque converter's efficiency. A compromise must be made between the magnitude of axial thrust and the torque converter's efficiency. Second, a portion of the trapped oil can be removed by introducing balancing holes, as shown in Figure 37. Third, the force  $F_3$  can be further reduced by reducing the overall surface area on which the centrifugal pressure acts. A smaller toroidal cross section (Figure 38) is implemented to meet this objective. A larger outer diameter would be required (to compensate for the reduction in overall surface area) if comparable performance characteristics are desired.

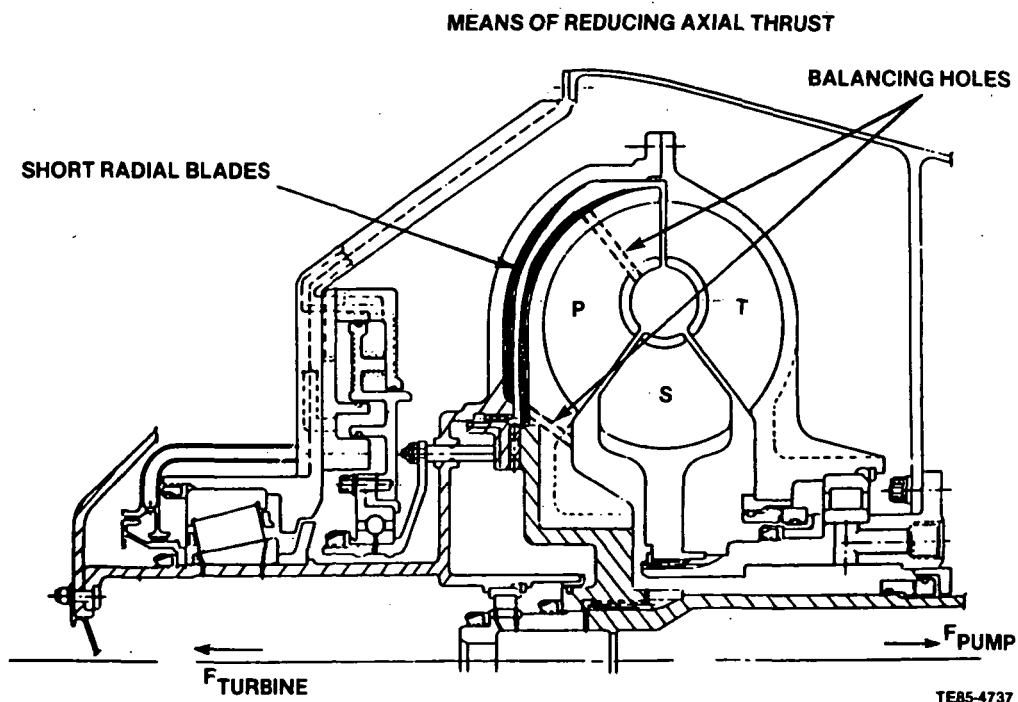


Figure 37. Axial thrust encountered within the torque converter means of reducing axial thrust.



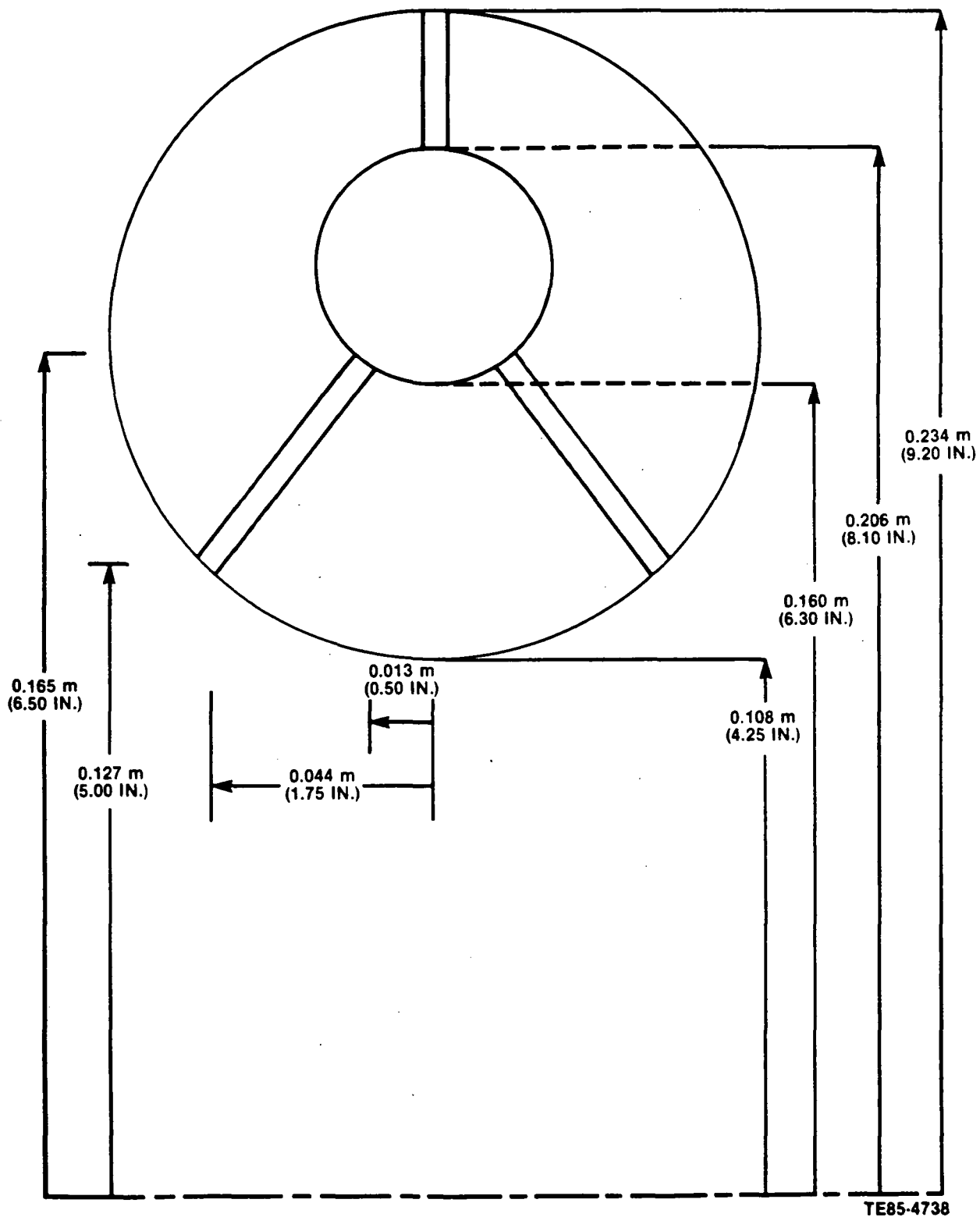


Figure 38. Modified toroidal cross section for minimum thrust with  $P_{\text{charging}} = 1379 \text{ kPa}$  ( $200 \text{ lb/in.}^2$ ).

Based on the available experimental data at the power and speed levels encountered in the automotive application, it is predicted that the axial forces  $F_{\text{pump}}$  and  $F_{\text{turbine}}$  would be reduced to about 44,482 N (10,000 lb) if the three concepts mentioned were incorporated into the design of the convertible engine torque converter. However, since no experimental data at the levels required for the convertible engine application are available, research and development effort must be undertaken to determine the effects of short radial blades, balancing holes, and smaller toroidal cross section on axial thrust and performance.

## CAVITATION

The torque converter's blade tip speed required for the convertible engine torque converter is more than twice that of converters used in the automotive application. There are two types of cavitation relating to the convertible engine torque converter:

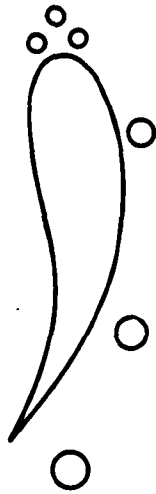
1. Cavitation associated with the inlet flow condition--During the first 6 sec of filling mode, the torque converter inlet charging pressure is essentially zero. Since the inlet charging pressure is less than the vapor pressure of the fluid, vapor bubbles are formed and may grow within the blade passage to relatively large sizes, as shown in Figure 39. These bubbles finally collapse downstream in a region of higher pressure. The repeated cycle of bubble formation and collapsing near solid surfaces may lead to cavitation erosion. A significant reduction in the performance of the torque converter can result, as indicated by Figure 15. This type of cavitation may be minimized by pressurizing the torque converter's cavity with the engine discharge compressor air prior to the filling mode.
2. Cavitation associated with poor hydrodynamic blade design--Figure 40 shows a typical pressure distribution around a pump blade element. The blade must be designed in a manner so that its minimum pressure is much higher than the vapor pressure of the fluid in order to avoid cavitation. If it was impossible to avoid the bubble formation, the torque converter pump element might be designed to operate under supercavitating conditions. Under these conditions, large size bubbles are formed, but bubble collapse occurs downstream of the pump blades (i.e., in the region between the pump exit and turbine inlet). It is important in a supercavitating torque converter that the bubbles should not collapse on the blade and that the blade passages should not choke.

In summary, cavitation is the main limitation to improvement in performance of all hydrodynamic machines. Additional work in the area of experimental and theoretical analysis of three-dimensional flow must be taken in order to significantly advance this technology.

## VIBRATION

Vibration in the convertible engine torque converter is an issue of concern for several reasons. First, the torque converter's blades can receive excitations from many sources, such as the following:

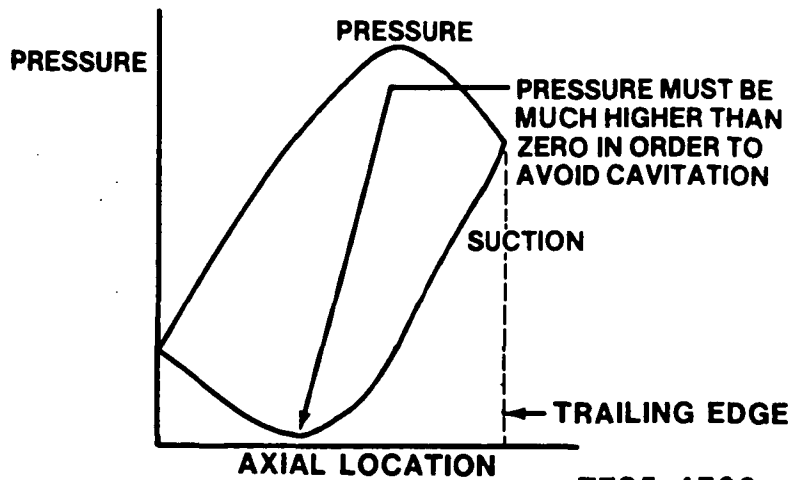
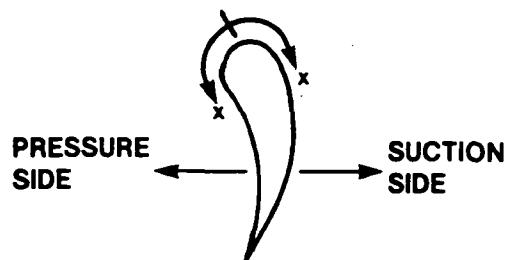
- o nonuniform flow distribution at the inlets and outlets



BUBBLES ARE FORMED BECAUSE THE INLET CHARGING PRESSURE IS ZERO. VAPOR BUBBLES MAY GROW WITHIN THE BLADE PASSAGES TO RELATIVELY LARGE SIZES.

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Figure 39. Cavitation associated with the inlet flow condition.



TE85-4739

Figure 40. Cavitation associated with poor hydrodynamic blade design.

- o nonuniform metal distribution at the inlets and outlets
- o nonaxisymmetric and nonsteady flow conditions

Second, many modes of vibration exist in all rotating turbomachines. It is therefore necessary to shift all the natural frequencies out of the range of operating speed. Third, the convertible engine torque converter is a dump and fill type of torque converter. It is therefore important to have uniform metal and flow distributions.

Vibration is a common problem in turbomachinery, and detailed research and technology effort to determine the actual loadings, mode shapes, and natural frequencies is required to solve this problem.

## IX. TORQUE/CONVERTER/CONVERTIBLE ENGINE INTEGRATION

The dump/fill torque converter is shown integrated into the engine envelope in Figure 41. The converter pump is driven by the LP turbine. The converter turbine drives the fan. The converter stator is grounded to static structure. Provisions to pressurize the torque converter cavity using compressor bleed air are included to reduce cavitation of oil during the torque converter fill sequence. The torque converter cavity also serves as a sump to collect oil exhausted by the torque converter during operation. The power takeoff shaft turns at a higher speed than the power turbine shaft to reduce shaft diameter and blockage of air entering the gas generator compressor. Diffuser passage length is increased as required to accommodate the power takeoff (PTO) shaft and drive gears in the location selected. An auxiliary air inlet is provided to ensure adequate flow area into the gas generator compressor inlet during operation in the turboshaft engine mode.

Axial load accountability for the No. 1, 2, and 3 bearings is shown in Figure 42 during vertical takeoff, transition, and cruise. The No. 1 tapered roller bearing absorbs the forward or positive (+) force of the fan. The No. 3 bearing absorbs the rearward or negative (-) force of the power turbine. The No. 2 bearing absorbs the positive force of the PTO drive load during the turboshaft engine operating mode. The normal loads on these bearings are shown during vertical takeoff and cruise when the torque converter is empty. During transition, or coupling and decoupling of the fan, peak reaction loads between the converter pump and turbine occur near lockup speed. The range of these loads during the 7-sec transition are shown for the transition (condition 2). Since these loads are of short duration and occur just twice per flight cycle, the engine life requirement can be achieved.

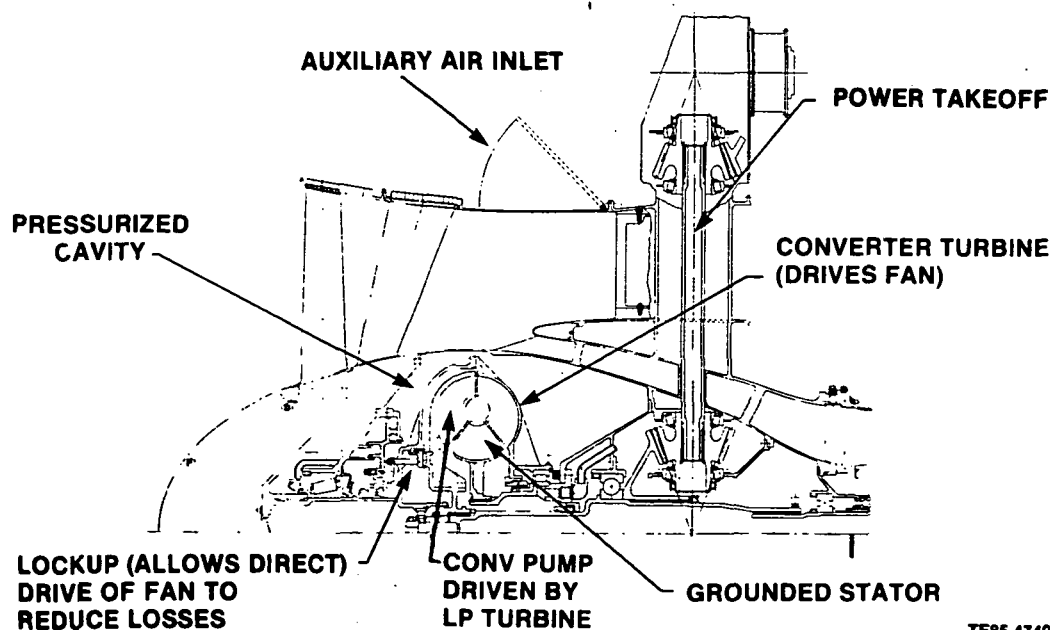


Figure 41. Torque converter integration.

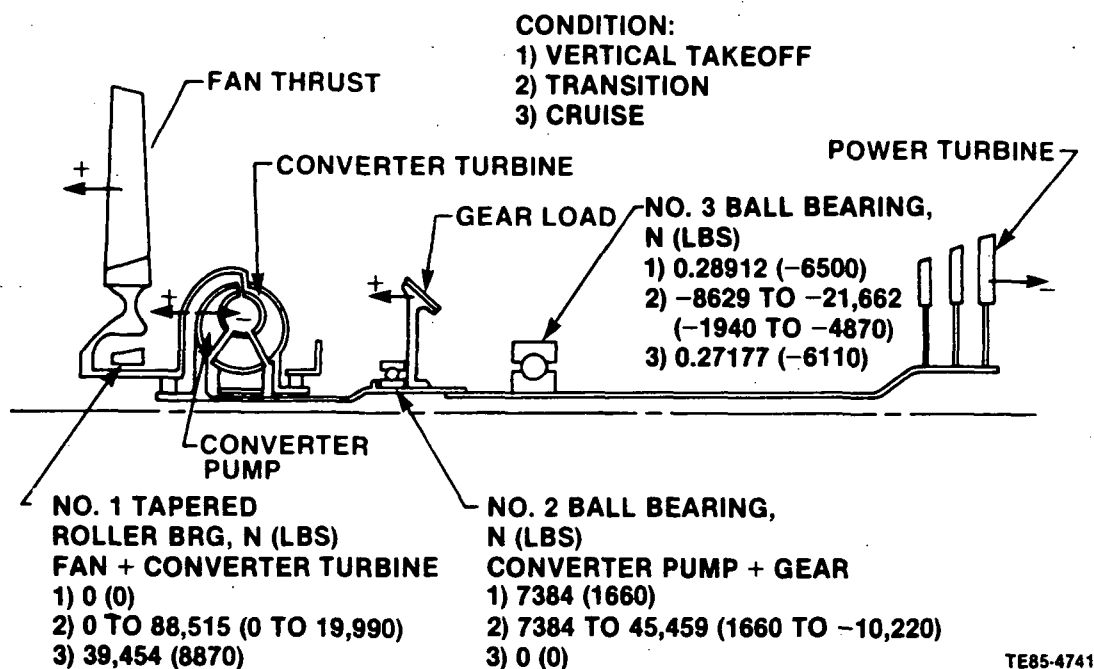
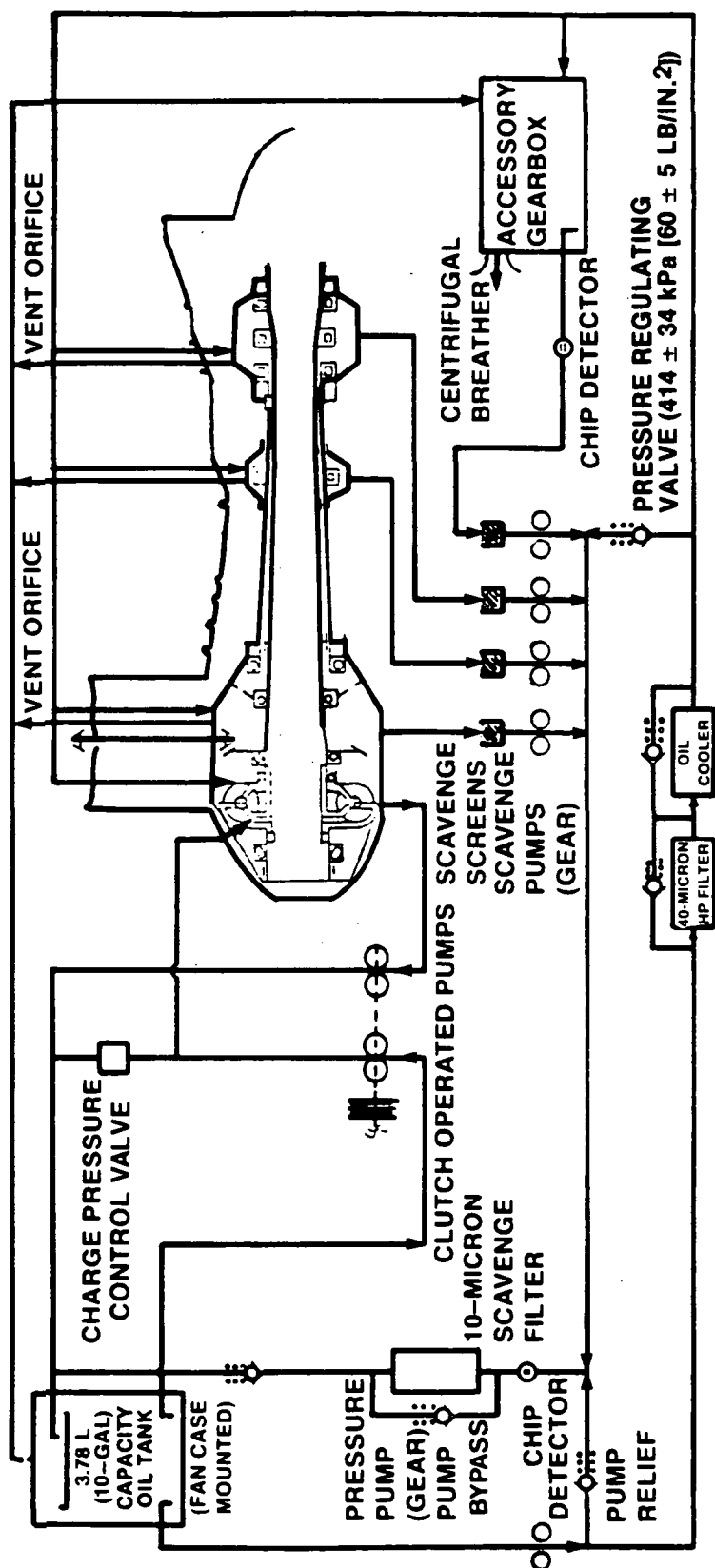


Figure 42. Thrust balance.

The integrated engine oil system schematic is shown in Figure 43. A 37.8 L (10 gal) oil tank supplies sufficient system capacity for the complete convertible engine including the 15.1 L (4 gal) capacity torque converter. The engine oil pressure pump supplies oil to the engine bearings and gears including the accessory gearbox and PTO and the torque converter bearings and shafts. Engine scavenge pumps return oil from the bearing gearbox and torque converter sumps. The torque converter cavity forms the sump for return of torque converter oil to the reservoir. Additional scavenge pump capacity is provided during charging of the torque converter by the combined charging oil pressure and scavenge pumps that are engine driven through an electric clutch.

A detailed weight summary for the convertible engine is shown in Table VI. Total engine weight is 641 kg (1413 lb) including 472 kg (1041 lb) for the fan and gas generator and 169 kg (372 lb) of additional items required for convertible engine operation. The torque converter weight with 15.1 L (4 gal) of oil is estimated at 88 kg (195 lb).



**Figure 43. Integrated oil system schematic.**

Table VI.  
Engine weight summary.

	<u>Weight--kg (lb)</u>
LP compressor--fan rotor, fan case, and support (composite)	130.0 (286)
HP compressor--inlet, rotor, and case	81.6 (180)
Diffuser and combustor	22.2 (49)
HP turbine--rotor and case	40.4 (89)
Transition	20.0 (44)
LP turbine--rotor, case, and rear support	112.9 (249)
Engine control and accessory drive	<u>65.3 (144)</u>
Total, fan and core	472.4 (1041)
Additional items:	
Propeller drives	14.5 (32)
Torque converter (includes 4 gal oil)	88.4 (195)
Added doors	18.1 (40)
Extended compressor inlet	6.8 (15)
Oil tank	11.3 (25)
Extra oil	20.4 (45)
Propeller drive gearbox	<u>9.1 (20)</u>
Sub total	168.6 (372)
Total weight	641.0 (1413)

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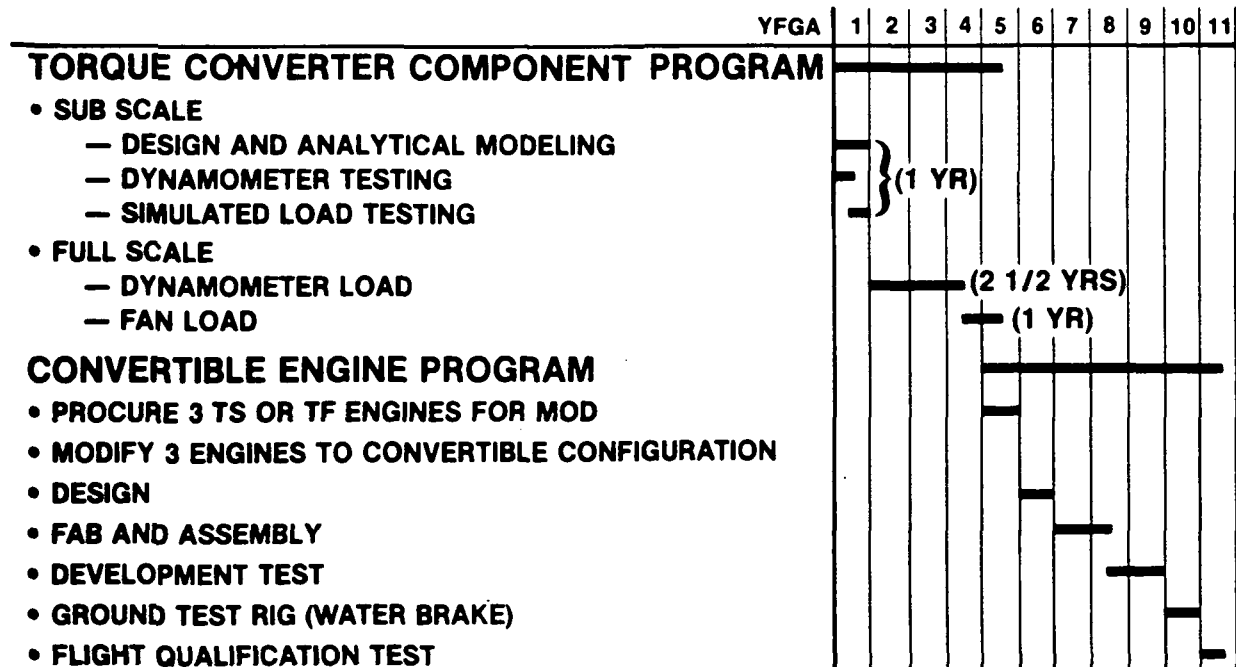
## X. WORK PLAN

The overdrive torque converter has good potential for convertible engine application; however, much higher torque converter pump and turbine tip speeds than those used in automotive applications are required, leading to potential problems with cavitation, erosion, and performance prediction. Charging pressure and cavity back pressure will be used to reduce cavitation. The dump/fill requirement may lead to problems in fluid unbalance. Uniform entry and exit of oil is required to avoid unbalance. High axial force may be diminished adequately by blades on the turbine hub to reduce static pressure. Other potential problems typical of turbine machinery, such as blade and vane vibration and forced response to engine order frequencies, may be encountered. Different numbers of blades in pump, turbine, and stator; uniform metal distribution on dynamically balanced rotating parts; and support tailoring to shift natural frequencies out of the operating range represent the approaches used in dealing with vibration problems.

The program plan shown in Figure 44 begins with test of the torque converter as a component and leads to a convertible engine program.

In the component program, an automotive size unit is tested first, then a 0.467 m (18.4 in.) diameter unit as described in the conceptual design. The objectives of the component program are as follows:

- o demonstrate performance over range of speed ratios
  - o full and partially filled
  - o range of charging pressures and cavity pressures to determine effect on cavitation



TE85-4743

Figure 44. Program plan.

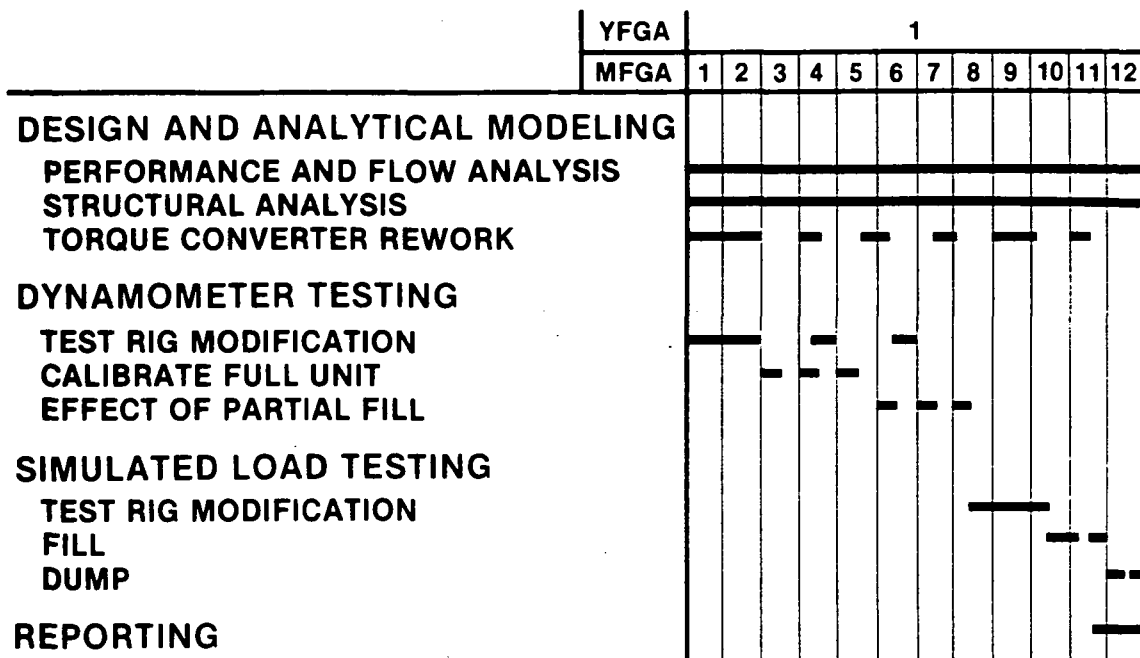
- o demonstrate dump/fill cycle with acceptable balance and vibration characteristics
- o obtain acceptable axial force level
- o investigate structural integrity and dynamics
- o demonstrate 1:1 speed ratio
- o demonstrate mechanical lockup feature
- o develop analytical modeling capability for torque converter design

The convertible engine program leading to flight qualification begins with modification of available turboshaft or turbofan engines to a convertible engine configuration using a torque converter. Development test is followed by test on a ground rig in which a water brake is used to load the engine PTO drive system and simulate the prop rotor load during the fan coupling and decoupling sequence.

The subscale component test is done with the baseline automotive torque converter modified to operate at higher speed, power, and torque than was tested on the original design, and to incorporate dump/fill provisions similar to the convertible engine design. The baseline automotive torque converter has a 0.292 m (11.5 in.) diameter and rotates at 2400 rpm for a tip speed of 36.5 m/s (120 ft/sec). This unit provided the experimental data base for the subject study of an overdrive, dump/fill torque converter. The new test would be done at 4000 rpm and 61 m/s (200 ft/sec) tip speed at an input torque of 217 N-m (160 lbf-ft). The purpose of the test is to provide initial insight into potential problems of fluid unbalance, axial thrust, and cavitation in an automotive size unit using available test facilities. The program is relatively inexpensive to accomplish, test units are available for modification, and setup changes and test part changes can be made with less expense.

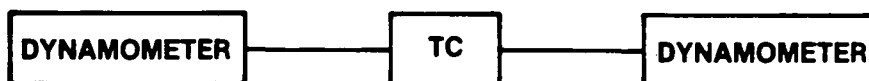
The subscale program is a one-year effort, as shown in Figure 45. Design and analytical modeling is continuous. Dynamometer testing in which the unit is calibrated full and partially filled under dynamometer load, as illustrated in Figure 46, is accomplished and includes time allocation for torque converter rework and test rig modification. Dynamometer testing under a fan load simulated by a water brake is also included. Details of simulated fan load test are shown in Figure 47.

The full-scale torque converter component program is done with the 0.467 m (18.4 in.) diameter unit as described in the conceptual design. The program involves the design and test of three torque converter units. The test units are fabricated from steel since fabrication with this material is satisfactory for tests up to lockup speed and is much cheaper than titanium. Units 1 and 2 are fabricated without lockup hardware since primary emphasis will initially be placed on torque converter performance. The program includes concurrent computer modeling for fluid flow, stress, and dynamics so that the computer models are updated as required after each test. Testing will include dynamometer rig testing using a large Allison industrial gas turbine engine for power supply with the torque converter loaded by a dynamometer. Testing will also include a simulated fan load using a modified turbofan engine fan. Figure 48 shows a schematic of the test setup with the dynamometer load and the fan load. The program leads to design of a flightweight unit.



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Figure 45. Subscale program.



CALIBRATE PERFORMANCE WITH FULL TC, 10 SPEED RATIOS, 2 CAVITY AIR PRESSURES, 10 CHARGING PRESSURES, AT VARIOUS LEVELS OF CONSTANT INPUT TORQUE (68-217 N-m [50-160 LBF-FT])

EFFECT OF PERCENT FILL AT SEVERAL SPEED RATIOS

TE85-4745

Figure 46. Standard steady-state performance testing.



DETERMINE PERFORMANCE FROM EMPTY TO FULL AT 3 FILL RATES, 10 SPEED RATIOS, 2 CAVITY AIR PRESSURES AT VARIOUS LEVELS OF CONSTANT INPUT SPEED (1000 - 4000 RPM)

DETERMINING PERFORMANCE FROM FULL TO EMPTY AT THE AVAILABLE DISCHARGE RATE, 10 SPEED RATIOS, 2 CAVITY PRESSURES AT VARIOUS LEVELS OF CONSTANT INPUT SPEED.

TE85-4746

Figure 47. Nonsteady-state dynamometer performance testing.

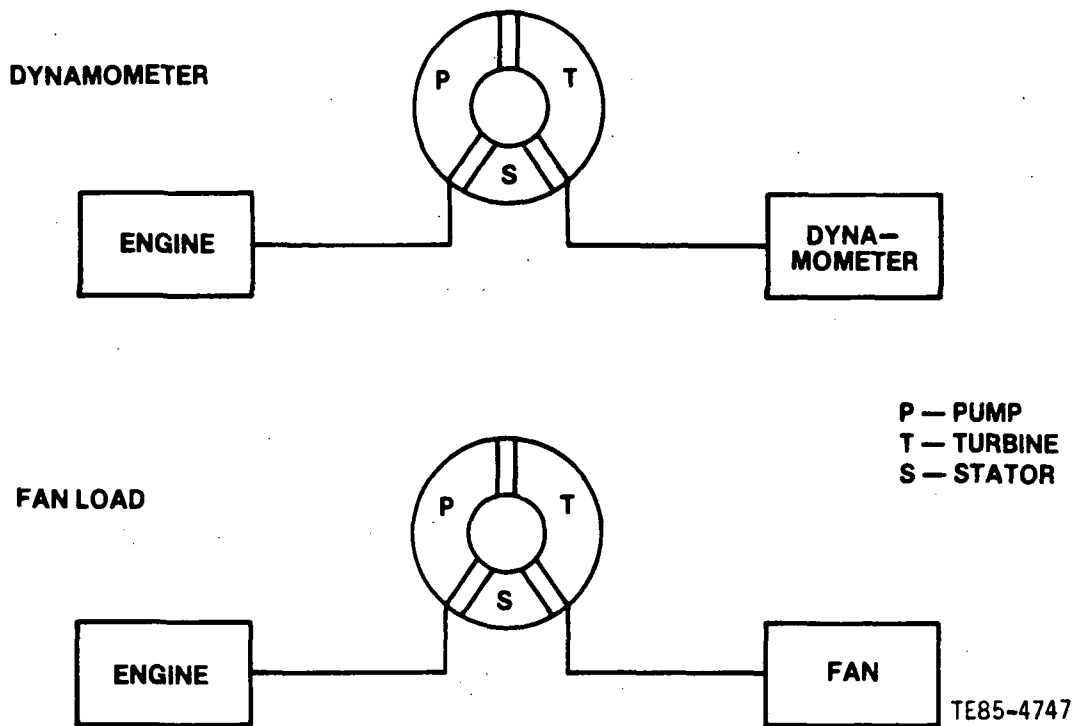


Figure 48. Experimental setup--full scale test.

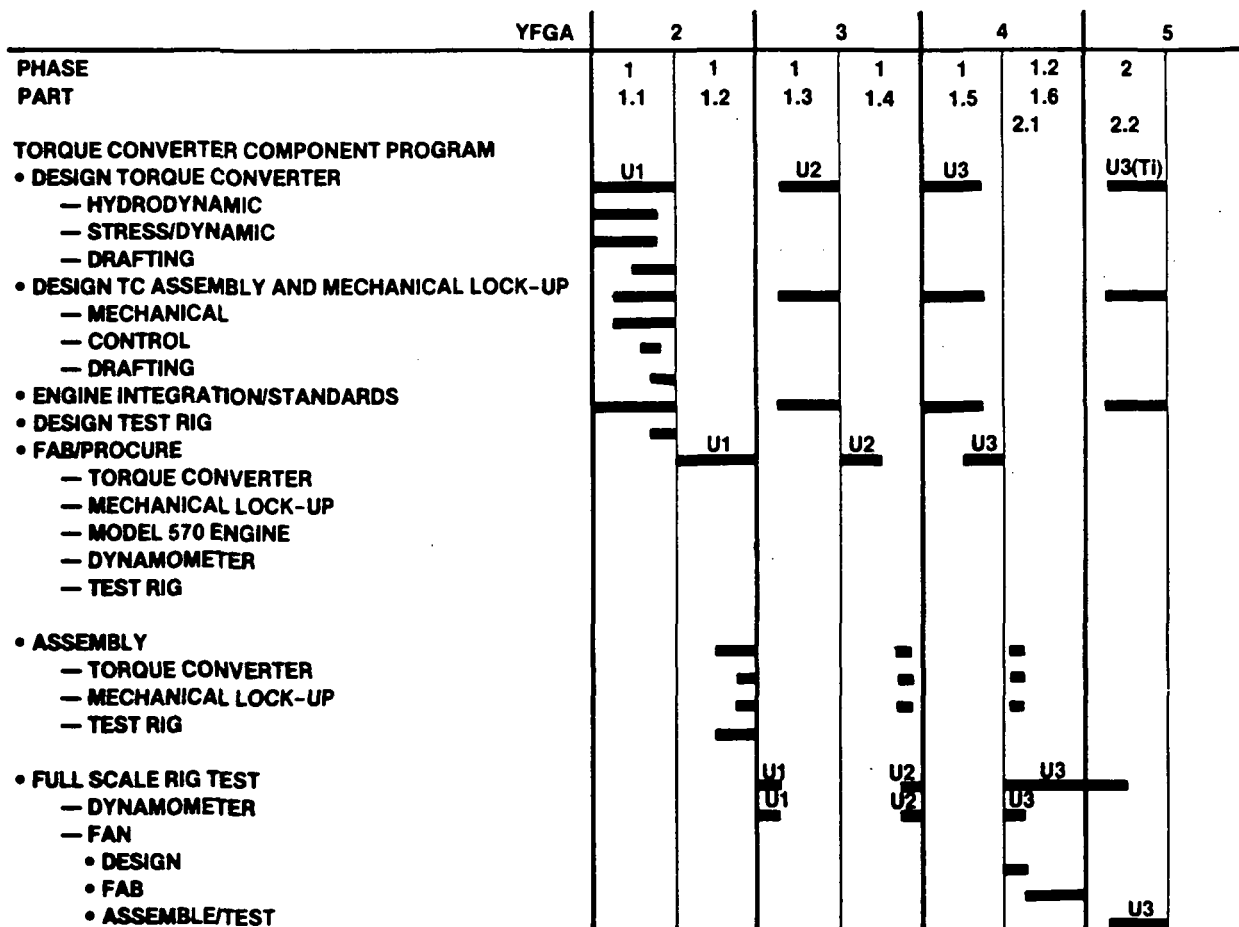
The workscope for the full-scale program is shown in Figure 49. Phase 1 covers testing of the torque converter with loading by a dynamometer on units 1, 2, and 3. Phase 2 is testing with the fan load on unit 3 that incorporates the mechanical lockup drive and design of a flightweight unit. The content of the full-scale program is as follows:

- | <u>Phase</u> | <u>Part</u>  |
|--------------|--|
| 1            | <p>1.1 Design and detail prototype unit 1 torque converter. The test rig design includes test housing and systems; enclosure, oil system, control system; test hardware; and drive couplings, support equipment, and instrumentation.</p> <p>1.2 Procure parts for unit 1 and test rig and assemble.</p> <p>1.3 Test unit 1. Update torque converter design computer models based on unit 1 test and design unit 2.</p> <p>1.4 Fabricate unit 2 (cast steel), assemble, and test.</p> <p>1.5 Design unit 3 based on test results of units 1 and 2 and update computer models. Fabricate unit 3 (cast steel, mechanical lockup).</p> <p>1.6 Assemble and test unit 3.</p> |
| 2            | <p>2.1 Design and procure test rig incorporating fan to load torque converter.</p> <p>2.2 Test unit 3 with fan load. Update design process and design flightweight unit No. 3.</p>   |

The Phase 1 dynamometer rig test is detailed as follows:

#### Test program

1. Calibrate performance with full torque converter (TC) at constant input torque of 5151 N-m (3800 lbf-ft), ten speed ratios, ten charging pressures, and three cavity pressures.
2. Determine performance from empty to full at three fill rates, ten speed ratios, and three cavity pressures at  $N_i = 7000$  rpm. Run each speed ratio at constant output speed and determine variation of torque versus time. Limit torque input to 5151 N-m (3800 lbf-ft).
3. Determine performance from full to empty at the available discharge rate, ten speed ratios, three cavity pressures at  $N_i = 7000$  rpm. Run each speed ratio at constant output speed and determine variation of torque versus time.
4. Provide data on heat rejection to oil during tests 1, 2, and 3.
5. Provide stress data at critical area during tests 1, 2, and 3.



TE85-4748

Figure 49. Torque converter component program (full scale).

6. Provide axial thrust data during tests 1, 2, and 3.
7. Provide windage loss data (empty TC) as available from tests 1, 2, and 3.
8. Test mechanical lockup operation in and out at no load and full load (unit 3 only).

The torque converter is tested at constant input torque of 5151 N-m (3800 lbf-ft) that can be supplied by the Allison Model 570 free turbine industrial engine. Torque converter performance over the full range of speed ratios can be mapped for a range of charging pressures and cavity pressures for the full torque converter. The transient dump/fill performance can also be measured up to 5151 N-m (3800 lbf-ft) input torque over a range of speed ratios as indicated. Although this procedure does not test the unit up to full torque, it does exceed the input torque used during fan acceleration in the convertible engine model, as shown in Figure 50. Data on heat rejection, stress, axial thrust, windage power, and mechanical lockup operation can also be obtained.

The Phase 2 fan load test is detailed as follows:

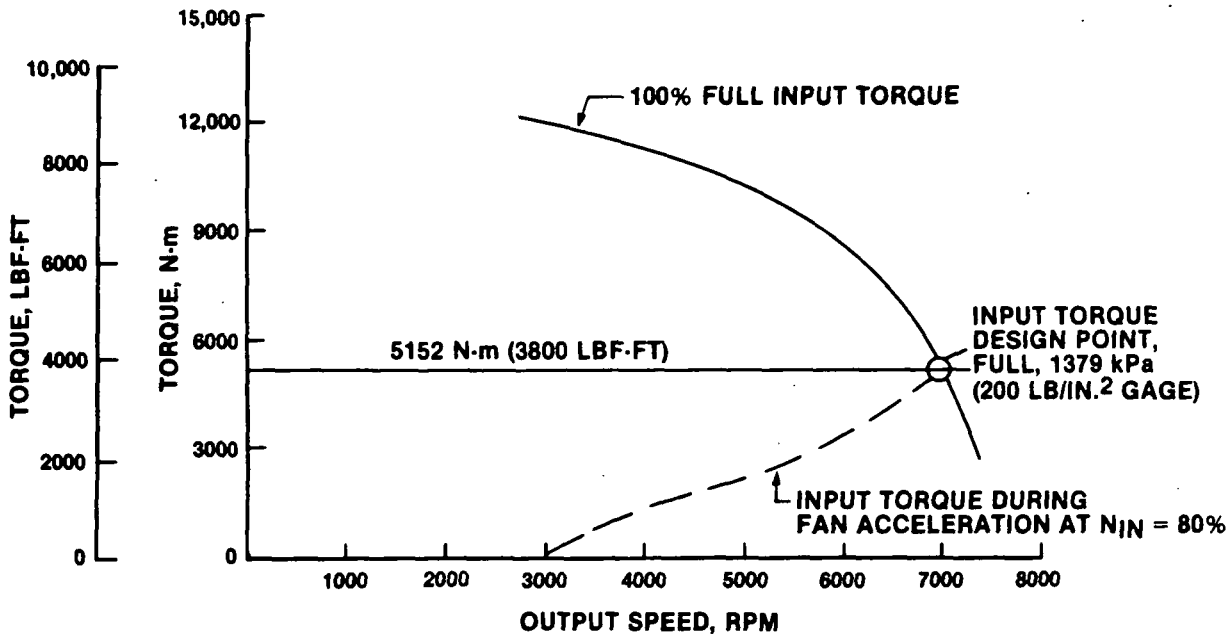
#### Test program

1. Couple fan at  $N_i = 7000$  rpm from  $N_{fan} = \text{windmill}$  using fill rates, charging pressure schedule, and cavity pressure selected from previous tests. Accelerate to maximum speed ratio.
2. Decouple fan at  $N_i = 7000$  rpm using charging pressure schedule and cavity pressure selected from previous tests.
3. Repeat 1 and 2 at several variants of above schedule.
4. Repeat 1 and explore effect of charging pressure on ability to synchronize input speed.
5. Couple fan as in 1 selected schedules and engage mechanical lockup and dump oil.
6. Fill torque converter with fan engaged as in item 5, release mechanical lockup, and decouple fan using selected schedules.
7. Run ten complete coupling and decoupling cycles simulating typical mission usage. Run at least 15 min in mechanical lockup mode before each decoupling. Run at least 15 min in decoupled mode before each coupling.

The objective of the test is to demonstrate coupling and decoupling of the fan and mechanical lockup using torque converter unit 3. The Model 570 engine again serves as the power source. The similarity of the modified fan from the Allison TF41 turbofan engine to the fan load and speed used in the feasibility study is shown in Figure 51.

Successful completion of the component test program will provide a suitable basis to design a flightweight torque converter for application to a convertible engine.

SL/370 km/h (200 KTAS)



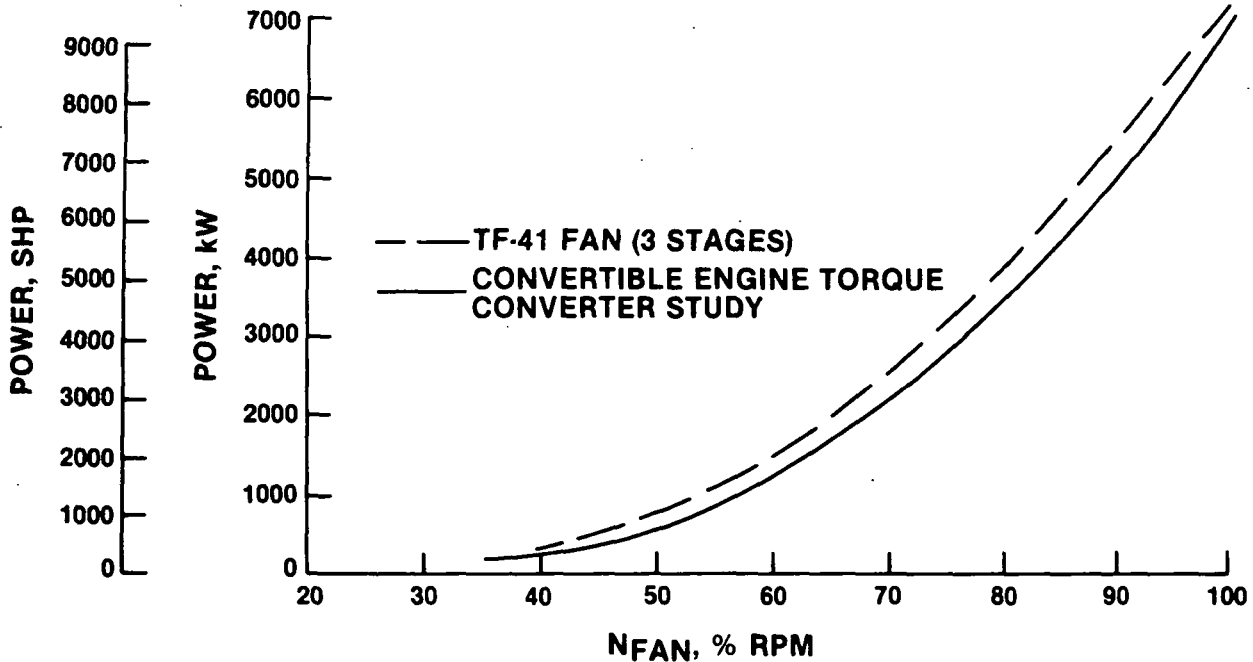
TE85-4749

Figure 50. Input torque used during fan acceleration.

SL/370 km/h (200 KTAS)

100%  $N_{FAN} = 8,734$  RPM (BASE)

100%  $N_{FAN} = 8,768$  RPM (TF-41)



TE85-4813

Figure 51. Fan power requirement.

## XI. RECOMMENDATIONS AND CONCLUSIONS

While underdrive torque converters with hydraulic clutches for mechanical lockup are in automotive use, an application in an aircraft gas turbine engine requires that new technical information be generated that is applicable to the special requirements imposed. Torque converters have been built and tested at normal power at size and speed encountered in automotive use but not at the power, diameter, and speed for convertible fan/shaft engine operation.

The central purpose of the torque converter feasibility study was to provide a conceptual design based on contemporary data and to recommend a technology program to bring the concept to a state of readiness for full-scale development.

The feasibility study has shown that the torque converter has excellent potential for turbofan/shaft engine use. The torque converter space requirements permit internal housing at acceptable system weight. The torque converter oil system can be integrated with the engine oil system without the need for additional heat exchanger capacity. The fan can be accelerated and mechanically locked to the power turbine shaft in less than 10 sec. The fan aerodynamic design is basically uncompromised for convertibility leading to an efficient and quiet cruise turbofan. Adequate shaft power is available from the engine for helicopter mode operation with minimal residual thrust from the engine.

However, these results are based on extrapolation of existing data on torque converters from automotive applications. Potential problems with the torque converter must be explored in test programs as recommended. Cavitation and its effects need to be explored. Means to reduce the high axial force potential must be demonstrated. Suitable designs to minimize fluid unbalance must be obtained.



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16. Abstract <p>The convertible engine torque converter provides a means to decouple the fan in a turbofan engine and thus make gas generator output available to turn a shaft. The resulting convertible engine is a turbofan/turboshaft engine capable of delivering thrust or shaft horsepower. Its application is for high speed rotorcraft such as the fold tilt rotor where shaft power is used to drive a propotor which can be tilted to provide rotor lift in the helicopter mode or propeller thrust in the wing borne flight mode. Turbofan engine thrust is needed for propulsion when the aircraft is in the high speed configuration with the proprotors folded. The convertible engine provides both functions at much less weight and installation complexity than separate turboshaft and turbofan engines.</p> <p>The objectives of the Feasibility Study for Convertible Engine Torque Converter program were to prepare a conceptual design of a gas turbine housed flight weight torque converter for primary use in a convertible fan shaft engine and to define a work plan to solve problems identified in the study that require research and technology effort.</p>					
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